

**THERMAL MANAGEMENT OF HIGH BRIGHT  
LED DISPLAY SYSTEM  
BY FORCED CONVECTION  
AND COOLING PERFORMANCE OPTIMIZATION  
WITH ANALYTICAL AND NUMERICAL APPROACHES**

A Thesis

by

Ahmet Salih Adalı

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Approved by:

---

Assoc. Prof. Özgür Ertunç, Advisor  
Dept. of Mechanical Eng.  
*Özyeğin University*

---

Prof. Bayram Şahin  
Dept. of Mechanical Eng.  
*Istanbul Technical University*

---

Asst. Prof. Altuğ Melik Başol  
Dept. of Mechanical Eng.  
*Özyeğin University*

Date Approved: August 13, 2024





*To my family*

## ABSTRACT

Thermal management of display devices is crucial for the long-term performance and life of the products. In line with customer demands, multipurpose display devices designed for operating in various environments must comply with high expectations of brightness. However, high brightness requirements mean considerable power consumption, leading to significant heat dissipation within the device. Analytical and numerical approaches are commonly used to optimize cooling performance by tailoring cooling system designs. The heat generation and dissipation inside the display systems could be modeled by using several numerical tools (e.g. ANSYS Icepak). The main objective of this research is to develop and optimize a cooling system that maintains the temperatures of critical components within safe operational limits, thereby keeping the overall reliability and efficiency of the display system. To achieve this, the study investigates various cooling methodologies while focusing on the combination of conduction and forced convection mechanisms to achieve the desired thermal management. Key findings show that a tailored cooling system, designed with a combination of conduction and forced convection, can significantly improve thermal management. The study proposes a design methodology that minimizes the number of components in the cooling system while allowing high cooling performance under harsh operating conditions. Comparison of analytical and numerical results and prototype testing validate the feasibility of the proposed cooling solutions. In summary, our study contributes valuable insights into the thermal management of high-bright LED displays. It offers practical solutions for indoor and semi-outdoor display applications. Future work will focus on further refining the cooling design and exploring advanced materials and technologies to enhance thermal performance.

## ÖZETÇE

Ekranlı cihazların termal yönetimi, ürünlerin uzun dönem performansı ve ömrü için oldukça önemlidir. Müşteri talepleri doğrultusunda, çeşitli ortamlarda çalışmak üzere tasarlanmış cihazlar, yüksek parlaklık beklentilerini karşılamalıdır. Fakat, yüksek parlaklık beklentileri hatırı sayılır bir güç tüketimi anlamına gelir ve bu da cihaz içinde önemli bir ısı yayılımına yol açar. Soğutma performansını optimize etmek için analitik ve nümerik yaklaşımlar genellikle soğutma sistemi tasarımlarını özelleştirerek kullanılır. Ekranlı cihazlardaki ısı üretimi ve dağılımı bilgisayar ortamında çeşitli nümerik araçlarla (ANSYS Icepak) modellenenir. Bu araştırmanın ana amacı, kritik bileşenlerin sıcaklıklarını güvenli çalışma sınırları içinde tutan ve böylece ekran sisteminin genel güvenilirliğini ve verimliliğini artıran bir soğutma sistemi geliştirmek ve optimize etmektir. Bunun için çalışma, istenen termal yönetimi sağlamak üzere iletim ve zorlanmış taşınım mekanizmalarının kombinasyonuna odaklanarak çeşitli soğutma yöntemlerini araştırır. Temel bulgular, iletim ve zorlanmış taşınım kombinasyonu ile tasarlanmış ve özelleştirilmiş bir soğutma sisteminin termal yönetimi önemli ölçüde iyileştirebileceğini göstermektedir. Çalışma, zorlu çalışma koşulları altında yüksek soğutma performansı sağlarken soğutma sistemindeki bileşen sayısının en aza indirilmesini hedefleyen bir tasarım metodolojisi önermektedir. Analitik ve nümerik sonuçların karşılaştırılması ve prototip testleri, önerilen soğutma çözümlerinin uygulanabilirliğini doğrulamaktadır. Özetle, yüksek parlaklıklı LED ekranların termal yönetimi için değerli bilgiler sağlanmaktadır. Hem iç mekan hem de yarı iç mekan ekranları için pratik çözümler sunar. Gelecekteki çalışmalar, soğutma tasarımını daha da iyileştirmeye ve termal performansı artırmak için gelişmiş malzemeler ve teknolojileri keşfetmeye odaklanacaktır.

## ACKNOWLEDGEMENTS

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# Nomenclature

## Greek and Roman Symbols

|               |                                                          |
|---------------|----------------------------------------------------------|
| $\delta_{ij}$ | Kronecker Delta                                          |
| $\epsilon$    | Rate of Dissipation of Turbulent Kinetic Energy          |
| $\mu$         | Dynamic Viscosity                                        |
| $\mu_t$       | Turbulent Eddy Viscosity                                 |
| $\nu$         | Kinematic Viscosity                                      |
| $\omega$      | Specific Rate of Dissipation of Turbulent Kinetic Energy |
| $\rho$        | Density                                                  |
| $\rho_0$      | Reference Density                                        |
| $\tau_{ij}$   | Viscous Stress Tensor                                    |
| $\tau_{ij}^R$ | Reynolds Stress Tensor                                   |
| $\varepsilon$ | Surface Roughness                                        |

## Other symbols

|             |                                  |
|-------------|----------------------------------|
| $\dot{Q}$   | Rate of Convection Heat Transfer |
| $\dot{q}_E$ | Source of Energy Per Unit Mass   |
| $A_s$       | Surface Area                     |
| $AC$        | Alternating Current              |
| $CFD$       | Computational Fluid Dynamics     |

*CFM* Cubic Feet per Minute

*CPU* Central Processing Unit

$D_h$  Hydraulic Diameter

$D_{eff}$  Effective Diameter

*DC* Direct Current

*DLED* Direct-lit Light Emitted Diode

$e$  Sum of Internal Energy and Kinetic Energy Per Unit Mass

*ELED* Edge-lit Light Emitted Diode

$f$  Friction Factor

$g$  Gravitational Acceleration

$h$  Heat Transfer Coefficient

*IC* Integrated Circuit

*IPA* Isopropyl Alcohol

*IR* Infrared

$K$  Minor Loss Coefficient

$k$  Thermal Conductivity

$L_c$  Critical Length

*LCD* Liquid Crystal Display

*LED* Light Emitted Diode

*MB* Main Board

*MOSFET* Metal Oxide Semiconductor Field Effect Transistor

*Nu* Nusselt Number

*OLED* Organic Light Emitted Diode

*P* Pressure

*p* Perimeter

*P<sub>f</sub>* Friction Losses

*P<sub>m</sub>* Minor Losses

*P<sub>tot</sub>* Total Losses

*PB* Power Board

*PCB* Printed Circuit Board

*Pr* Prandtl Number

*RANS* Reynolds Averaged Navier-Stokes

*Re<sub>L</sub>* Reynolds Number

*RGB* Red Green Blue

*rpm* Revolutions per Minute

*S<sub>ij</sub>* Strain Rate Tensor

*T* Temperature

*T<sub>0</sub>* Reference Temperature

*T<sub>∞</sub>* Ambient Temperature

*T<sub>f</sub>* Film Temperature

$T_s$     Surface Temperature

$UV$     Ultraviolet

$V$     Velocity





# CHAPTER I

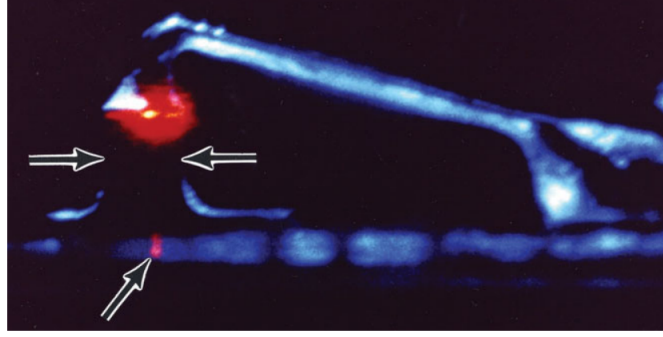
## INTRODUCTION

### *1.1 Motivation of the Study*

The need for thermal management of high-bright display applications in line with customer demands drives the motivation for this study. As the display industry continues to evolve, there is a significant demand for higher quality and energy-efficient displays. This study explores the cooling system of high bright light-emitting diode (LED) TVs, focusing on their design, functionality, and performance to drive innovation and meet consumer expectations. Also, effective thermal management is crucial for maintaining the long life and efficiency of these devices. This research seeks to support manufacturers in improving product reliability by providing comprehensive insights.

### *1.2 Literature Review*

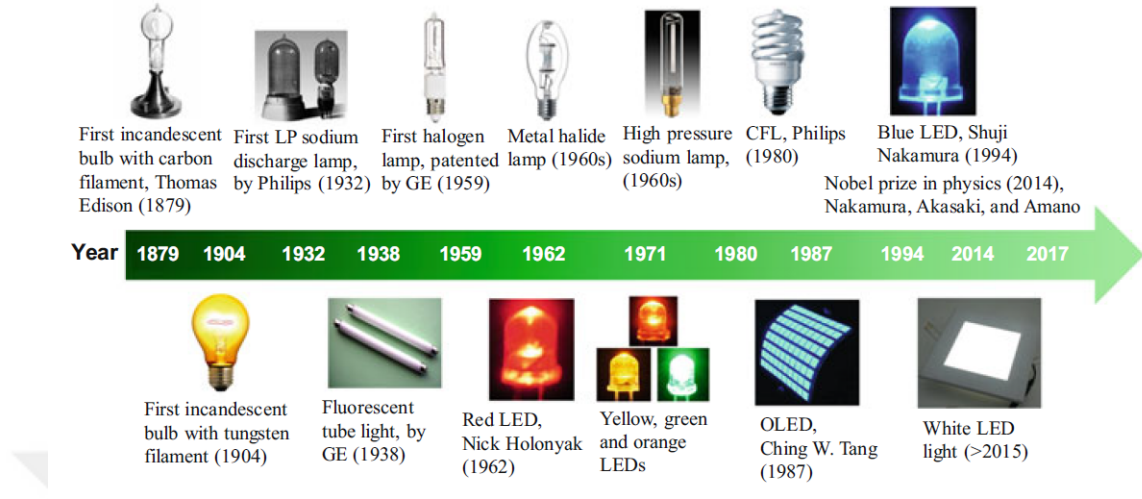
LED display devices are breakthrough devices in the development of display technology. Conceptually, LEDs were first created at the beginning of the 1960s. Nick Holonyak Jr. developed the first visible-spectrum LED in 1962. This allows future advancements and transformations in LED technology, from basic indicator lights to functional components in modern electronic devices. LEDs have significantly improved in brightness, efficiency, and color gamut. Shuji Nakamura's invention of blue LEDs in the 1990s made it possible to produce white and red-green-blue (RGB) LEDs, which are essential for full-color displays [5].



**Figure 1:** The first visible spectrum led developed by Nick Holonyak Jr. [1].

The significance of thermal management in LED display devices cannot be overestimated. High-brightness LEDs produce a lot of heat, negatively impacting the life, performance, and efficiency of the product. Efficient thermal management is crucial for preventing overheating, maintaining steady operation, and guaranteeing the life span of LED displays [6]. The need for large-scale, high-brightness screens is growing, leading to the need to solve thermal issues.

In the 1960s, constant advancements and developments happened in LED display technology. In the first stages, because of their narrow color gamut and luminous efficacy, LEDs were only valuable in low-brightness applications. However, significant developments in LED materials, including gallium arsenide phosphide (GaAsP), made it possible to produce red, yellow, and green LEDs in the 1970s and 1980s [7]. LEDs are now more widely used in digital displays, calculators, and other electronic devices.



**Figure 2:** Major milestones in LED development [2].

The development of blue LEDs in the 1990s was game-changing since it made white and full-color screens possible. This innovation had a crucial role in expanding the uses of LEDs, which resulted in their use in large-scale advertising displays, computer monitors, and televisions [8]. LEDs are now the favored option for various display applications due to recent advances in LED technology that have improved brightness, energy efficiency, and color accuracy [5].

Research and development have been the driving force behind the advancement of LED technology. The manufacturing techniques and semiconductor materials have evolved to a point where LEDs now have much higher luminous efficacy (lumens per watt). Improvements in multi-chip LEDs and phosphor coatings have also enhanced brightness and color rendering. Additionally, flexible and transparent displays have been made possible by developing organic LEDs (OLEDs), increasing the potential applications of LED technology [9].

LED displays can be used in semi-outdoor, outdoor, and indoor environments.

The term indoor environment describes areas of a building where people spend a significant amount of time, usually up to 90% of the day. These areas are used for residential, business, educational, and medical purposes. Indoor environment quality

is essential for comfort, productivity, and human health [10]. Digital signage, video walls, computer monitors, and televisions frequently employ indoor LED displays. These display devices are designed for controlled light environments with vivid colors and high resolution. However, efficient thermal management is required to avoid overheating, particularly in high-brightness applications.

Semi-outdoor environments have partial enclosure and outdoor exposure, presenting particular difficulties in determining thermal comfort. Some examples of semi-outdoor environments include bus terminals, atriums, storefronts, shopping malls, lobbies, and balconies [10]. Semi-outdoor LED displays are utilized in semi-outdoor environments. These devices are weather-resistant and brighter than indoor displays to prevent ambient light. Balancing brightness with power consumption and thermal management is challenging for semi-outdoor displays.

In outdoor environments, display devices are subjected to direct sunlight, wind, rain, and temperature changes, which require extremely durable and bright LED technology. These displays must remain highly readable and visible regardless of the weather. Typical features include waterproof enclosures, ultraviolet (UV)-resistant coatings, and sophisticated thermal management systems that employ liquid cooling and fans as active cooling techniques to maintain ideal operating temperatures [10].

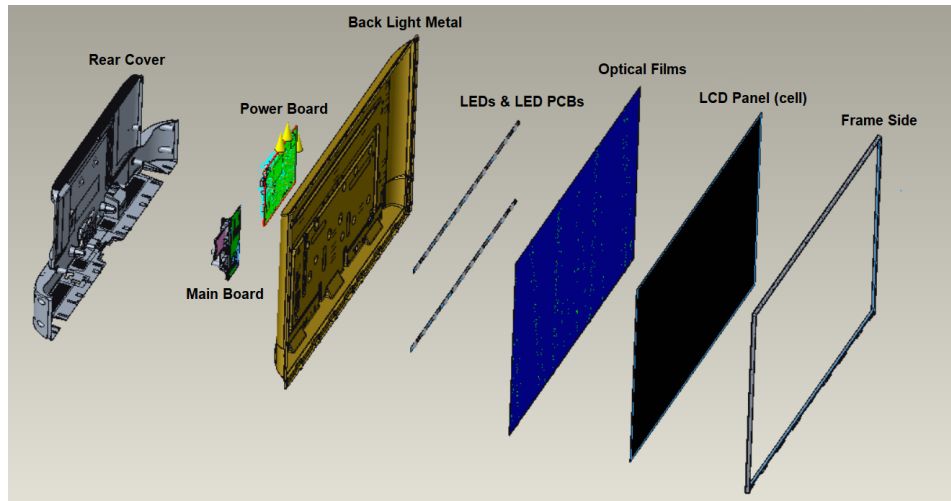
The use of LED displays has increased dramatically over time. LEDs are currently utilized in various applications, including automotive lighting, medical equipment, architectural lighting, and more conventional uses like televisions and computer monitors. Due to its energy efficiency, durability, and versatility, LED technology is preferable for many applications [11].

LED displays have many benefits, such as extended operating life, excellent energy efficiency, and durability. LEDs require less power than conventional light sources like incandescent and fluorescent lights, making them an energy-efficient choice for display applications. Furthermore, thanks to their long life, there is less need for regular

replacements, which reduces the maintenance costs [11]. However, LED screens have some limitations. High-brightness LEDs produce a lot of heat; therefore, thermal control is critical to sustaining long life and performance. Color shift can occur in LEDs over time, affecting display uniformity. In addition, LED displays may initially cost more than conventional lighting options, but this is frequently compensated for by their longer lifespan and lower operational costs [7].

Controlling the heat produced by high-bright LEDs is one of the significant issues facing the development of LED displays. Excessive heat can result in thermal deterioration, decreased efficiency, and a shortened lifespan. Effective heat control is necessary to avoid these problems and guarantee the steady operation of LED displays. Various thermal management strategies have been developed to solve these issues, including active cooling techniques like fans and liquid cooling systems and passive cooling techniques like heat sinks.

The main components of the LED display devices are liquid crystal display (LCD) panels, mainboards (MB), power boards (PB), and backlight units (BLU), including LEDs, LED printed circuit boards (PCB), and optical film. Figure 3 shows the basic components of the LED display device.



**Figure 3:** Basic components of the LED display device.

The LCD panel, commonly known as the **cell**, is a thin-film transistor (TFT) connected to an electronic driving gate circuit. The cell uses the light from the LEDs and the backlight unit to show images on the screen. It is one of the most expensive parts of the LED TV. Mainboards have crucial components such as the primary central processing unit (CPU), memory, graphics, and memory chips, as well as several interface connections required for the LED display device to function. Mainboard components are assembled using surface mounting technology, guaranteeing dependable and effective performance. Mainboard layout and design are critical for the LED display device performance metrics, including contrast, response time, and color saturation. The powerboard is essential for controlling and allocating power to the numerous parts of LED display devices. It transforms alternating current (AC) power into the proper direct current (DC) voltages for the internal parts of the LED display devices. The powerboard design must be flexible enough to cover a range of LCD panel sizes and layouts. Due to the increases in voltage and current output requirements, the design of the powerboards becomes increasingly complex as the panel size increases. Powerboards are also designed to consider the electrical needs of high-performance graphics chips and other critical components on the mainboards [12].

The backlight unit converts the point light source of LEDs into a uniform surface light source through diffuse reflection. Two main types of backlight units represent the position of the light source: direct-lit backlight (DLED) and edge-lit type (ELED) backlight. To enhance brightness, the backlight unit needs to include multiple optical films [13]. The brightness of the backlight is critical for indoor, semi-outdoor, and outdoor usage. Luminance or brightness is defined as the luminous flux per area per solid angle ( $\text{lm}/\text{m}^2/\text{sr}$  or nits). The homogeneity and brightness level of the LED backlight are crucial for its performance [14].

Consumers demand a display device for indoor display and indoor display with outdoor brightness applications. They desired to use this device in their house and

semi-outdoor areas (see Figure 4). To meet this demand, products must be designed for high operating brightness levels. To catch high brightness levels, the product must consume a high power. This means that heat dissipation of the power-consuming components, such as PB, LEDs, and MB, must be managed. The temperature inside the device must be kept under a specific temperature with a test condition of 40°C 90% of relative humidity due to reliability issues and requirements in the standards [15]. Maximum temperatures of the components must be under a specific limit determined by the manufacturer. The product is designed and served in the field. However, thermal management optimization must be studied to ensure the long life of the product, its efficiency, and power consumption.



**Figure 4:** High-bright LED TV.

### ***1.3 Open Fields in the Literature***

Many cooling methods are used to remove heat from heat sources. The cooling techniques generally used for electronic cooling are natural convection cooling and forced convection cooling. There are many other methods, such as synthetic jet cooling and phase change cooling [16] [17]. Several scientific works have been made on the thermal management of LED TVs. Generally, this is done using different materials

with different thermal conductivities coupled with natural convection. In one study, different materials and designs were used to create heat spreaders to remove heat from ELED and DLED TVs by natural convection [18]. Unlike general thermal management of display devices, some studies used forced convection to achieve thermal management. Another study examined an outdoor digital signage product (called a totem). By using open-loop and closed-loop fan structures, thermal management is achieved in this study under the worst outdoor conditions [19]. Chung assesses the effects of various channel geometries on pressure loss and heat transfer efficiency, including tapered designs and the addition of guide vanes in his study. The study finds that a tapered channel with guide vanes is an excellent passive cooling solution for outdoor display devices because it increases mass flow while lowering the maximum surface temperature, resulting in a considerable heat transfer improvement through numerical models and experimental measurements [20].

In the present study, the product which we try to optimize will be used for indoor and semi-outdoor applications. On sunny days, the brightness of objects outdoors can exceed 10000 nits, and most of the conventional display devices have brightness between 300-500 nits [21]. That is why a higher brightness level is required for outdoor and semi-outdoor applications than for conventional indoor display devices. The brightness level of 3000 nits and the DLED panel structure are chosen for market competition and hardware limitations for the product. The product requires high power consumption, and the heat created by the system cannot be managed by only natural convection. Therefore, to ensure the desired product reliability level, combining the conduction and convection heat transfer methods is crucial. The study aims to determine the optimal thermal management method concerning temperature limits to maintain the long life and performance of the components (integrated circuits (IC), LEDs, PCBs, Cells, etc.). It will also determine the most efficient way to cool the current system. The number of components in the cooling system will be optimized



and reduced by creating channels to cool critical regions (heat spots) of the current product. Analytical and numerical approaches, supported by thermal mappings, will be used.

#### ***1.4 Hypothesis of the Study***

The literature describes different natural and forced convection methods for managing the thermal problems in display devices. None of the studies in the literature focus on the thermal management of high-bright indoor and semi-outdoor display devices. Furthermore, the studies do not focus on component-scale thermal management. By strategically focusing the cooling system on critical components of the display device without changing the cooling elements and material properties used, it is possible to maintain the same cooling performance as the current system while reducing the overall number of cooling elements.

#### ***1.5 Objective of the Study***

The main objective of the study is to develop a methodology by controlling the temperature of the cell, powerboard, mainboard, and LEDs of the display devices under certain temperature limits in specifications while operating an environment at 40 °C to provide the desired functions in line with customer demand. This involves examining the thermal properties of the system, considering the working environment, and choosing the right cooling components, materials, and technologies. Hereafter, it is called a tailored cooling system in this thesis. The main intention is to design a tailored cooling system for semi-outdoor use. Before this thesis, the temperature measurement results for the system components showed that the current system meets the reliability limits for regular indoor use more than the requirements. The cooling system of the current display device was developed through experience-based techniques, practical procedures, prompt solutions based on empirical data, and intuitive decisions. Hereafter, it is called a heuristic cooling system in this thesis. Analytical

and numerical results are compared while developing the methodology before creating a prototype for testing. In addition, data collected from the critical regions of the test setup were compared with analytical and numerical results to ensure accuracy and reliability.

## **1.6 Methodology**

First, background information about analytical and numerical approaches is given, and the experimental approach is explained. The equations to be used in the analytical approach, the models used in the simulation studies, the measurement equipment for the experimental study, and the points to be measured are detailed. Analytical calculations of the product with a heuristic cooling system were performed, and the fan speeds required for cooling the system were determined. Then, the pressure drops of the product with a tailored cooling system were calculated, and the necessary optimum fan operating conditions were determined analytically. Then, a steady-state Reynolds-averaged Navier–Stokes (RANS) simulation was performed using Ansys Icepak, and the fan speeds in the simulation environment were obtained for the heuristic and tailored cooling systems. In addition, the critical temperatures of the system components, such as cell, MB, and PB, were determined. A mesh independence study was conducted to demonstrate the accuracy of the simulations. The test cases performed in the simulation environment were also examined experimentally. Measurements were taken on the components of the MB and PB and the cell surface via thermocouples and IR cameras from the measurement points determined in the simulation environment. The fan speeds were compared analytically and numerically, and their differences were interpreted. Afterward, the numerical and experimental temperature measurement results were compared. Finally, the experimental measurement results were compared, and the final findings were evaluated. The areas that can be examined and focused on for future studies were mentioned.

## ***1.7 Structure of the Thesis***

Chapter 1 includes background information about LED technology and LED display device developments, the hypothesis, the objectives of the study, and the methodology. Chapter 2 presents the analytical background of the study. Chapter 3 describes the numerical approach, the experimental setups, and the equipment used to measure the temperature of the products and their cooling systems. Chapter 4 gives the analytical, numerical, and experimental results of the study and provides a comparison between them. Chapter 5 discusses the key findings in the thesis, the limitations of the study, further investigation opportunities, and the similarities and differences of the findings from previous works in the literature. Finally, the conclusions are at the end of the thesis, as in Chapter 6.

## CHAPTER II

### ANALYTICAL BACKGROUND

#### *2.1 Analytical Approach*

Construction of the analytical approach requires understanding the continuity, momentum, and energy equations. In our problem, the fluid is Newtonian, and the flow is incompressible and steady, with constant viscosity and density. Temperature variations in the system will occur at different locations, and Boussinesq approximation is needed for the model construction. According to this approximation, the density variations are negligible. However, the density is considered when multiplied by gravitational acceleration (buoyancy term). This allows the simplification of the conservation equations. Boussinesq approximation assumes incompressible fluid with constant density in momentum and continuity equations, but body force is due to density variation [22]. The following equation describes the Boussinesq approximation.

$$\rho = \rho_0 [1 - \alpha(T - T_0)] \quad (1)$$

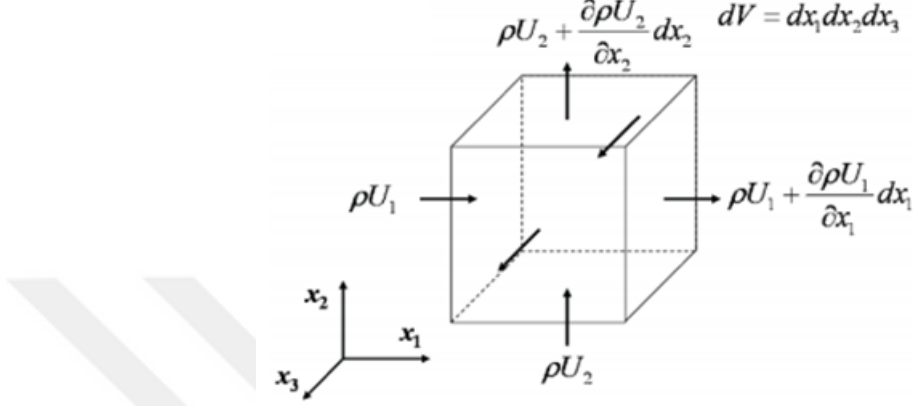
$\rho_0$  is the reference density at  $T=T_0$  and  $T_0$  is the reference temperature somewhere in the system.  $\alpha$  is the coefficient of thermal expansion.

Furthermore, the Boussinesq hypothesis is required for turbulence modeling. On the other hand, under the steady-state system assumption, steady-state heat conduction through the solid components is modeled with Fourier's law of heat conduction [23]. According to the Fourier law, the power consumption and thermal conductivities of the components are sufficient for the simulation inputs for the steady heat conduction problems.

Figure 5 represents the differential form of the continuity equation with tensor

notation.

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho U_i)}{\partial x_i} = 0 \text{ (where, } i = 1, 2, 3) \quad (2)$$



**Figure 5:** Differential control volume and the mass flux through its surfaces.

The first term is the rate of change of mass within a control volume, and it describes the accumulation that must be balanced. The second term is the net flow rate through the control surface. In this study, the flow is incompressible, and the flow is steady. Thus, the differential form of the continuity equation becomes

$$\frac{\partial U_i}{\partial x_i} = 0 \text{ (where, } i = 1, 2, 3) \quad (3)$$

The differential form of momentum equation with tensor notation is,

$$\rho \frac{\partial U_i}{\partial t} + \rho U_j \frac{\partial U_i}{\partial x_j} = \rho g_i - \frac{\partial P}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} \text{ (where, } i \& j = 1, 2, 3) \quad (4)$$

For a Newtonian fluid, the viscous shear is a linear function of the velocity gradient. Thus, the momentum equation becomes the Navier-Stokes equation.

$$\rho \left[ \frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} \right] = - \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \frac{2}{3} \delta_{ij} \mu \frac{\partial U_k}{\partial x_k} \right] + \rho g_i \quad (5)$$

(where,  $i \& j \& k = 1, 2, 3$ )

With incompressible and constant viscosity assumptions. Navier Stokes equation becomes,

$$\rho \frac{\partial U_i}{\partial t} + \rho U_j \frac{\partial U_i}{\partial x_j} = \rho g_i - \frac{\partial P}{\partial x_i} + \mu \frac{\partial^2 U_i}{\partial x_j^2} \text{ (where, } i \& j = 1, 2, 3) \quad (6)$$

Osbourne Reynolds worked on the Navier Stokes equations and derived Reynolds averaged Navier Stokes (RANS) equations. Reynolds suggested the decomposition of the flow variables in continuity and momentum equations into mean and fluctuating components. After the derivation of the equations, Reynolds stresses are appeared. Reynolds stresses lead to the closure problem, which is essentially the result of more unknown terms than equations in the equation system. At this point, turbulence models are needed to solve the closure problem. The Boussinesq hypothesis is used for this purpose, and it states that the dominating factor for momentum transfer in turbulent flow is large turbulent eddies [24] [25].

After the derivation of continuity and the momentum equation by using Reynolds decomposition, the equations become,

$$\frac{\partial \overline{U_i}}{\partial x_i} = 0 \text{ (where, } i = 1, 2, 3) \quad (7)$$

$$\frac{\partial \overline{U_i}}{\partial t} + \overline{U_j} \frac{\partial \overline{U_i}}{\partial x_j} = -\frac{1}{\rho} \frac{\partial \overline{P}}{\partial x_i} + \nu \frac{\partial^2 \overline{U_i}}{\partial x_j^2} + \frac{1}{\rho} \frac{\partial \tau_{ij}^R}{\partial x_j} \text{ (where, } i \& j = 1, 2, 3) \quad (8)$$

The term  $\tau_{ij}^R$  represents the Reynolds stress tensor. With Boussinesq hypothesis, the Reynolds stress tensor can be written as,

$$\tau_{ij}^R = 2\mu_t S_{ij} - \frac{2}{3} \rho k \delta_{ij} \text{ (where, } i \& j = 1, 2, 3) \quad (9)$$

The  $S_{ij}$  is the strain rate tensor and can be written as,

$$S_{ij} = \frac{1}{2} \left( \frac{\partial \overline{U_i}}{\partial x_j} + \frac{\partial \overline{U_j}}{\partial x_i} \right) \text{ (where, } i \& j = 1, 2, 3) \quad (10)$$

After the terms  $\tau_{ij}^R$  and  $S_{ij}$  placed to the momentum equation the momentum equation becomes,

$$\frac{\partial \overline{U_i}}{\partial t} + \overline{U_j} \frac{\partial \overline{U_i}}{\partial x_j} = -\frac{1}{\rho} \left[ \frac{(\partial \overline{P} + \partial^2 \rho k)}{\partial x_i} \right] + \frac{\partial}{\partial x_j} \left[ \frac{1}{\rho} (\mu + \mu_t) \frac{\partial \overline{U_i}}{\partial x_j} \right] \text{ (where, } i \& j = 1, 2, 3) \quad (11)$$

$\mu_t$  is the turbulent eddy viscosity. Turbulent eddy viscosity must be calculated to overcome the Reynolds closure problem. At this point, turbulence models need to be implemented. In the simulations, the two-equation model ( $k$ - $\epsilon$ ) is used, and turbulent eddy viscosity is defined as,

$$\mu_t = c_\mu \frac{k^2}{\epsilon} \quad (12)$$

$k$  is the turbulent kinetic energy and  $\epsilon$  is the the rate of dissipation of turbulent kinetic energy.

The differential form of energy equation with tensor notation is,

$$\frac{\partial \rho e}{\partial t} + \frac{\partial \rho e U_i}{\partial x_i} = \rho U_i g_i - \frac{\partial P U_i}{\partial x_i} + \frac{\partial U_i \tau_{ij}}{\partial x_j} + k \frac{\partial^2 T}{\partial x_i^2} + \dot{q}_E \quad (\text{where, } i \& j = 1, 2, 3) \quad (13)$$

The above equations are solved using the computational fluid dynamics (CFD) software. The shape of a surface, as well as its direction, influences natural and forced convection heat transfers. It is also affected by temperature variations on the surface and the fluid's thermophysical characteristics.

The two-equation model ( $k$ - $\epsilon$ ) was selected for the problem, and the reason for this selection is mainly the computational time gain for the wall function assumption instead of near-wall treatment. It allows mesh reduction in the problem compared to other turbulence models such as  $k$ - $\omega$  [26].

In this study, radiation effects are considered in CFD simulations. The ray-tracing model is used as a radiation model. In this model, ray paths are calculated and stored before the fluid flow calculations. Complex geometries require a lot of computational time. To reduce computational time, surfaces and cells are clustered into surface and volume "clusters" to minimize the number of radiating surfaces and absorbing cells. The incident radiation flux and volume sources are calculated for the surface and volume clusters. The surface and volume cluster temperatures are calculated by area and volume averaging [27]. The ray tracing model is selected due to its accuracy and

computational time gain compared to the surface-to-surface radiation model.

### 2.1.1 Semi-Empirical Equations for External Flows with Heat Transfer and Internal Flow Losses

Many dimensionless numbers are used to solve the forced convection mechanism. In this study, the following correlations were implemented to understand whether the cooling performance of the heuristic cooling system is enough or not. The product with a heuristic cooling system is modeled with the assumption of horizontal and vertical plates, and the flow over these plates is assumed to be external. For this reason, external flow correlations are used to create a model. Reynolds number is essential to determining the forced convection flow regime, representing the ratio of inertial forces to viscous forces within a fluid [3].

$$Re_L = \frac{VL_c}{\nu} \quad (14)$$

The simple empirical correlation for the average Nusselt number in forced convection is,

$$Nu = \frac{hL_c}{k} = CRe_L^m Pr^n \quad (15)$$

Where m and n are constant exponents, and the value of the constant C depends on geometry and flow. All fluid properties are to be evaluated at the film temperature with the formula below.

$$T_f = \frac{T_s + T_\infty}{2} \quad (16)$$

The current system is modeled with vertical and horizontal plates. The characteristic length of the horizontal surface can be calculated using Equation 17 due to its importance for Nusselt number correlation calculations.

$$L_c = \frac{A_s}{P} \quad (17)$$



Using a basic relationship for convection heat transfer in Equation 18, the required heat transfer coefficient for maintaining the surface temperatures set for the components of the system can be found.

$$\dot{Q} = hA_s(T_s - T_\infty) \quad (18)$$

After obtaining the heat transfer coefficient, the Nusselt number is found to determine the flow characteristics. Equation 19 and Equation 20 represent the average Nusselt number over the flat plate for the external flow.

$$\text{For Laminar flow} \rightarrow Nu = \frac{hL}{k} = 0,664Re_L^{0,5}Pr^{1/3} \quad Re_L < 5 \times 10^5 \quad (19)$$

$$\begin{aligned} \text{For Turbulent flow} \rightarrow Nu = \frac{hL}{k} &= 0,037Re_L^{0,8}Pr^{1/3} \\ 0,6 \leq Pr \leq 60 \ \& \ 5 \times 10^5 \leq Re_L \leq 10^7 \end{aligned} \quad (20)$$

By following Equation 19 and Equation 20 with the help of Equations 14, fan velocities are calculated to maintain the predetermined surface temperature of components of the heuristic cooling system.

The fan channel system is created following basic design methodologies and numerical studies for the tailored cooling system. Using this radial fan channel structure, the operating point of the fans can be determined. Subsequently, the system and fan curves are compared to identify the optimum operating conditions. For this purpose, the internal flow correlations are implemented to calculate the system losses. The hydraulic diameter of the duct must first be determined using the equation below.

$$D_h = \frac{4A}{P} \quad (21)$$

Equation 22 is used to determine the system's effective diameter.

$$D_{eff} = \frac{64}{fRe_{D_h}}D_h \quad (22)$$

To find the loss of the system, the friction factor must be determined using Moody's Chart. The Reynolds number and relative roughness are required to determine the friction factor of the system.

$$\text{Relative roughness} = \frac{\varepsilon}{D_{eff}} \quad (23)$$

The total pressure drop of the system can be calculated by combining friction losses and minor losses using Equation 24.

$$\Delta P_{tot} = \Delta P_f + \sum \Delta P_m = \rho g \left( \frac{V^2}{2g} \frac{fL}{D_{eff}} + \sum K \frac{V^2}{2g} \right) \quad (24)$$

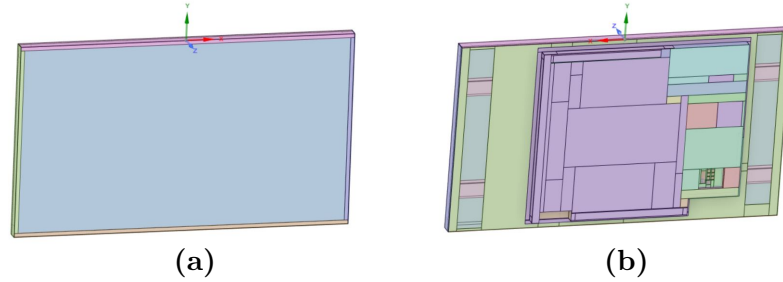
For different volumetric flow rates of the fan, the system loss curve can be created and combined with the fan curve for comparison.

## CHAPTER III

### NUMERICAL AND EXPERIMENTAL SET-UPS

#### 3.1 *Numerical Setup*

A numerical approach is employed to verify the analytical and experimental results of the systems. The present cooling system was created using heuristic approaches, and a tailored cooling system was developed using analytical methods, design rules, and limitations. ANSYS Icepak is used to perform finite element simulations for mentioned systems. The simulation uses the RANS method due to its effect on solution time and its application simplification. The two-equation turbulence model ( $k-\epsilon$ ) is used as a turbulence model. The model is simplified for the simulation to reduce the computational time and prevent mesh-caused problems like skewness and face alignment. Figure 6 shows the simplified model of the high bright LED display device for the simulation.

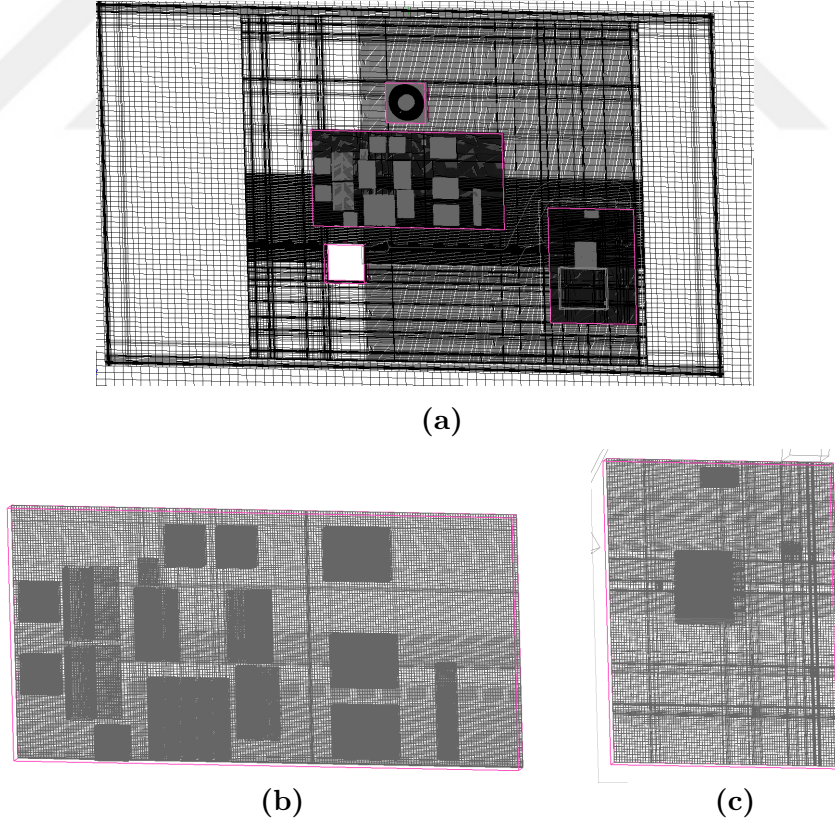


**Figure 6:** The front (a) and back (b) views of the simplified model of the high-bright LED display device for the simulations.

The current system is a heat-dissipating system with electronic boards and components on the product. This system is considered a steady-state system, and the simulation settings are set according to the steady-state thermal problem. The main

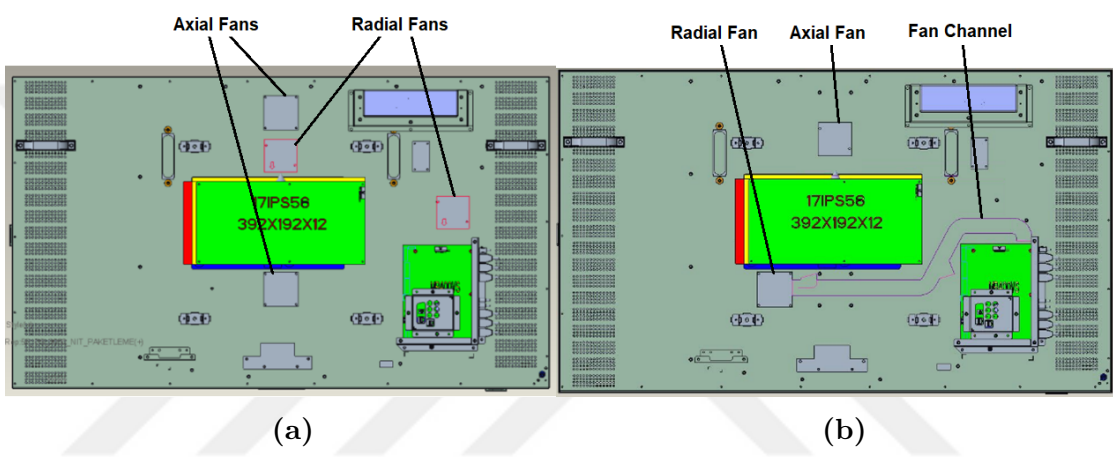
intention is to control the temperatures in the system when the heat-generated components reach their maximum temperatures. Time-dependent temperature changes are not focused. By following the steady-state assumption for the system, the simulation time was shortened compared to the transient thermal assumption.

Meshing is another crucial step for the simulations. To reduce the complexity of the problem and the number of meshes in the system, assembly meshing, which is called non-conformal meshing, is used for critical components like MB, PB, and LEDs to prevent mesh bleeding. Although non-conformal meshing has some disadvantages, it allows for a reduction in problem size and computational time. For this purpose, non-conformal mesh refinement is applied in the simulations [28]. Figure 7 shows the mesh structure of the product.



**Figure 7:** Cut plane view of the mesh structure of the product (a) and assembly mesh of the PB (b), and MB (c).

In addition, a mesh independence study must be carried out to ensure the accuracy of the numerical simulations. The current products with a heuristic and tailored cooling system are simulated before the experiments. The tailored cooling system design is required for optimized fan channel design to cool the critical components on MB and PB. Figure 8 shows the different cooling systems and components of the products.



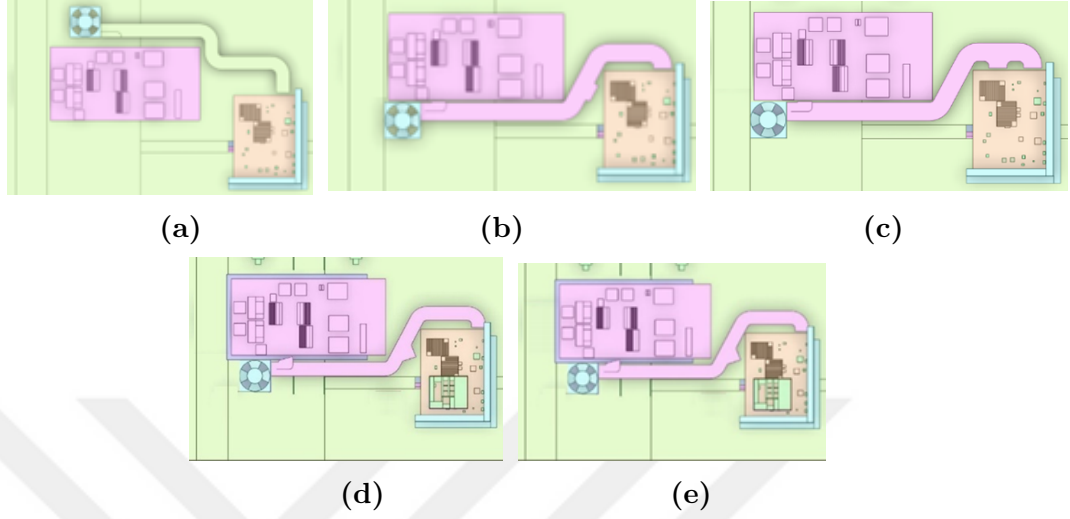
**Figure 8:** Different cooling systems and components of the products. (a) Heuristic cooling system and its components. (b) Tailored cooling system and its components.

### 3.1.1 Numerical Approach for Fan Channel Structure

This study aims to develop an efficient cooling system by keeping the cooling system components low. In the system, one axial and one radial fan is canceled. However, canceling one radial fan causes a cooling problem on the MB or PB side of the system. Every component on the boards doesn't need to be cooled down. Thus, a channel for radial fan to reach the critical components on the MB and PB is proposed to be developed.

During the development of the fan channel structure, available space inside the product, cabling, keep-out areas, and other structural constraints were considered. A couple of candidate designs were developed by following basic design methodologies and the design limitations of the current system. Figure 9 represents the candidate

channel designs before numerical simulations.

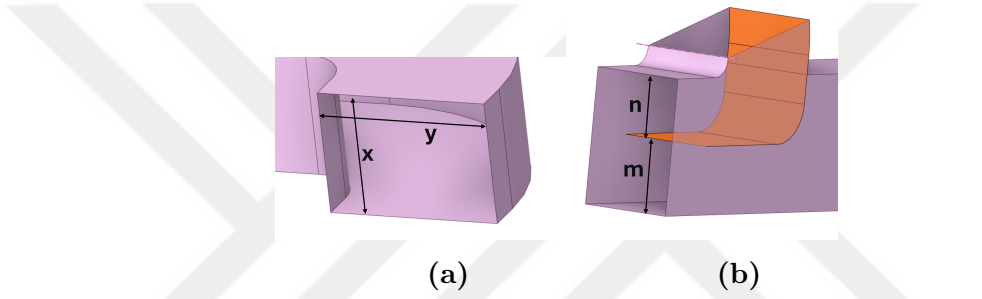


**Figure 9:** Different channel design options for cooling system.

Design (a) features a single radial fan pushing air through a long, curved channel connecting two separate system sections. The design is created by considering the available space inside the product and cablings. In design (b), the fan is positioned on the bottom side of the PB. The channel design has been changed in terms of cross-section and channel length. The bends on the channel are revised to reduce losses. One more opening is added to the MB side to increase its cooling performance. This design aims to provide more uniform cooling across the components. Design (c) shows a slightly different opening structure on the MB side of the system to optimize the airflow for specific areas on the MB. It focuses on cooling certain critical areas. In design (d), the fan channel is revised to directly target the MB and PB with a shorter channel compared to design (b) and design (c). Openings on the MB and PB sides have also been revised to direct the flow to the target components. This allows a more focused cooling approach where the airflow is concentrated on specific areas rather than distributed over surfaces. The design (e) evolves from design (d) by changing the position of the flow separator in the fan channel to adjust the flow rate that needs to be distributed to cool down the critical components on MB and PB. It provides

even more focused cooling to key components.

Table 1 shows the basic structural differences between candidate channel designs for the simulations. Where  $x$  and  $y$  represent the cross-section dimensions of the channel, and  $n$  and  $m$  represent the dimensions that define the position of the flow separator within the channel. The flow separator is a structural element placed inside the channel to direct the flow towards a specific point. For all candidate designs, this flow separator directs the flow to PB. Figure 10 describes the  $x, y, m$ , and  $n$  dimensions.



**Figure 10:** Critical fan channel dimensions. (a) Cross-section dimensions of the fan channel. (b) The dimensions for positioning the flow separator inside the fan channel.

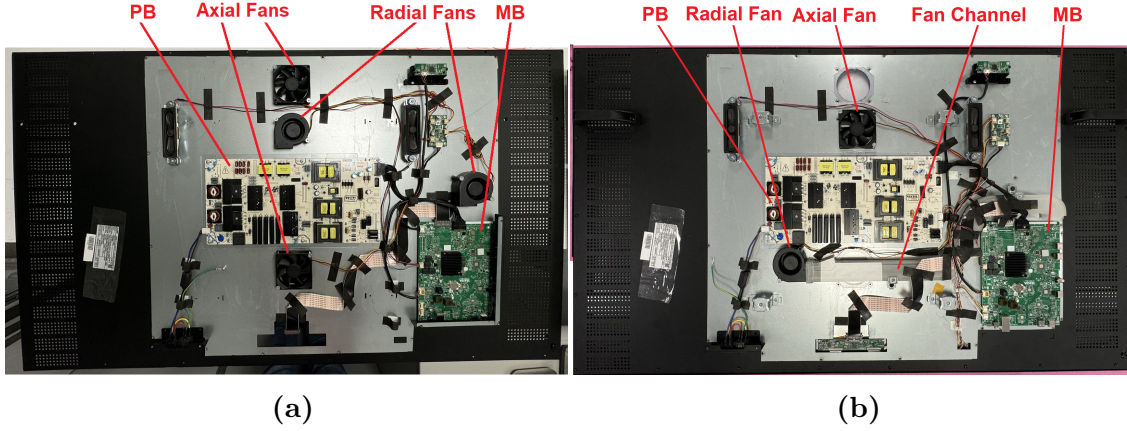
**Table 1:** The structural differences between candidate fan channel designs.

| Design Options | Channel Length (mm) | Cross Section |          | # of Bends | # of Openings at MB side | # of Openings at PB side | Flow Separator Positions at PB side |          |
|----------------|---------------------|---------------|----------|------------|--------------------------|--------------------------|-------------------------------------|----------|
|                |                     | $x$ (mm)      | $y$ (mm) |            |                          |                          | $n$ (mm)                            | $m$ (mm) |
| a              | 604                 | 30            | 35       | 3          | 1                        | 1                        | 17.5                                | 17.5     |
| b              | 630                 | 25            | 35       | 3          | 2                        | 1                        | 17.5                                | 17.5     |
| c              | 630                 | 30            | 40       | 3          | 2                        | 1                        | 20                                  | 20       |
| d              | 604                 | 25            | 35       | 3          | 2                        | 1                        | 17.5                                | 17.5     |
| e              | 604                 | 25            | 35       | 3          | 2                        | 1                        | 10                                  | 25       |

### 3.2 Experimental Setup

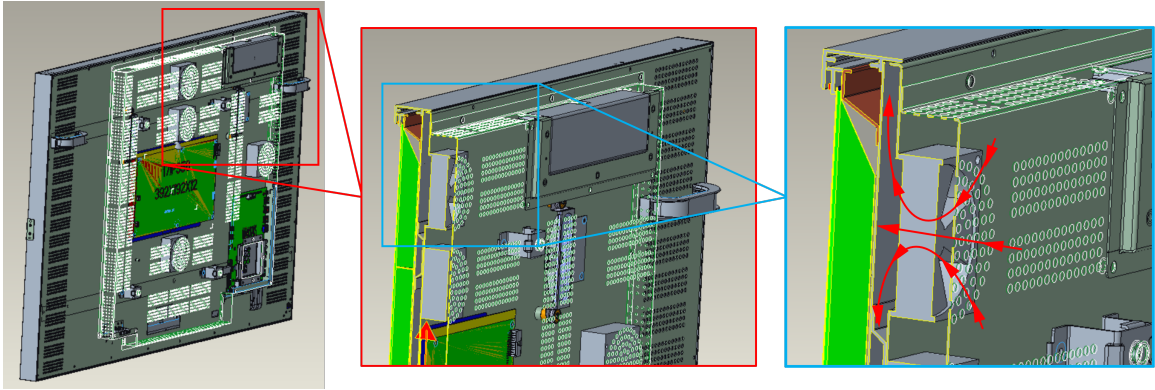
In the experiments, two different cooling design options were tested. Figure 11 shows the component placements and differences between heuristic and tailored cooling systems.





**Figure 11:** The back view of the products with heuristic and tailored cooling system. (a) Heuristic cooling system and component placements. (b) Tailored cooling system and component placements.

Axial fans blow the air from the ambient to the inside of the product to cool the LEDs and cell surface. Figure 12 represents the blowing direction of the axial fans for products with heuristic and tailored cooling systems.

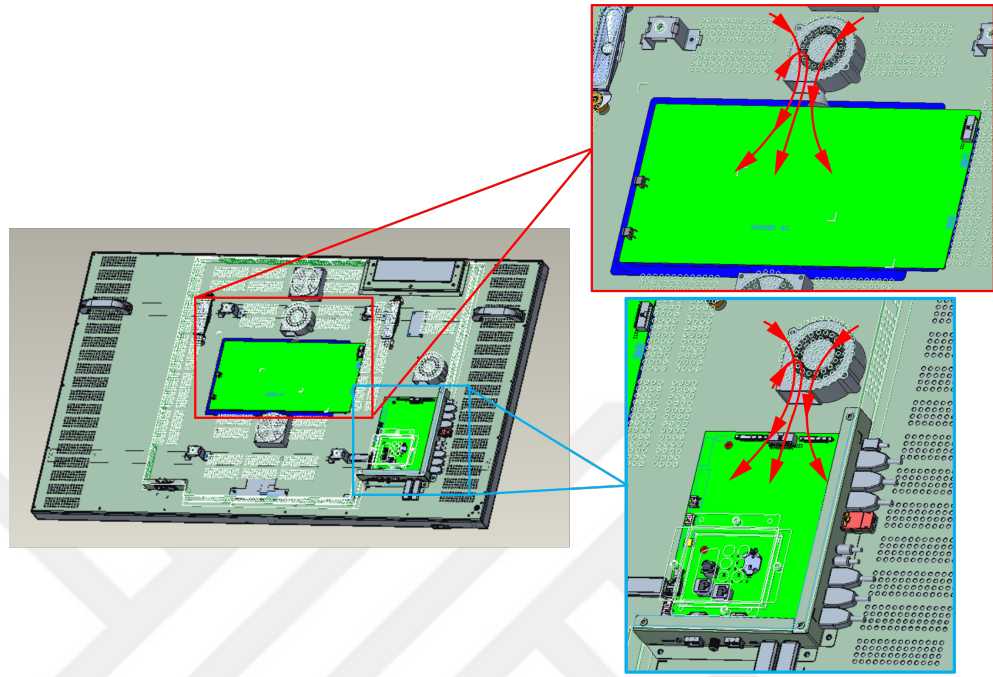


**Figure 12:** Flow direction sketch of the axial fan for the product with heuristic and tailored cooling systems.

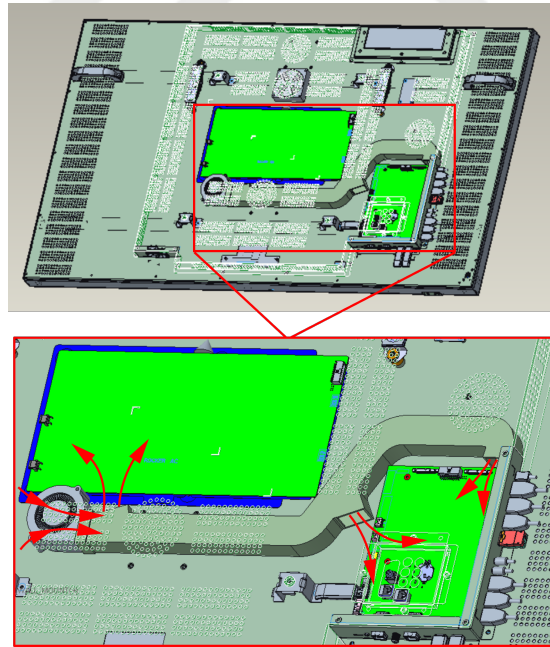
Radial fans blow air from the ambient to the boards in the product with a heuristic cooling system. The flow is precisely directed by the fan channel in the product with the tailored cooling system, ensuring efficient cooling. Figure 13 represents the flow



direction in both products.



(a)



(b)

**Figure 13:** Flow direction sketches of the radial fans for the product with heuristic (a) and tailored (b) cooling systems.

With the tests to be carried out, the surface temperatures of the cell, MB, and PB elements will be read at the maximum brightness level of the product, and it will be possible to compare the obtained values with numerical results. This comparison helps us understand the temperature levels of the tailored and heuristic cooling system design.

Thermocouples and Infrared (IR) cameras are used to check the temperatures of the desired points in the system.

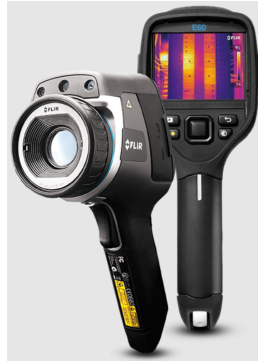
Thermocouples are precise measurement devices that determine the critical temperatures of heat-generated systems. They are connected to data loggers for temperature readings. Many types of thermocouples are available in the market, including K, J, and T types. The main difference between these types of thermocouples is the material of the wires [29]. This study uses K-type thermocouples due to their wide temperature measurement span, low prices, and availability in the market [30]. The system has 30 points for temperature measurements using thermocouples. Multichannel data loggers are required to read all measurements. Figure 14 shows the data logger used for temperature readings.



**Figure 14:** HIOKI LR8401 brand multichannel data logger for thermocouple readings.

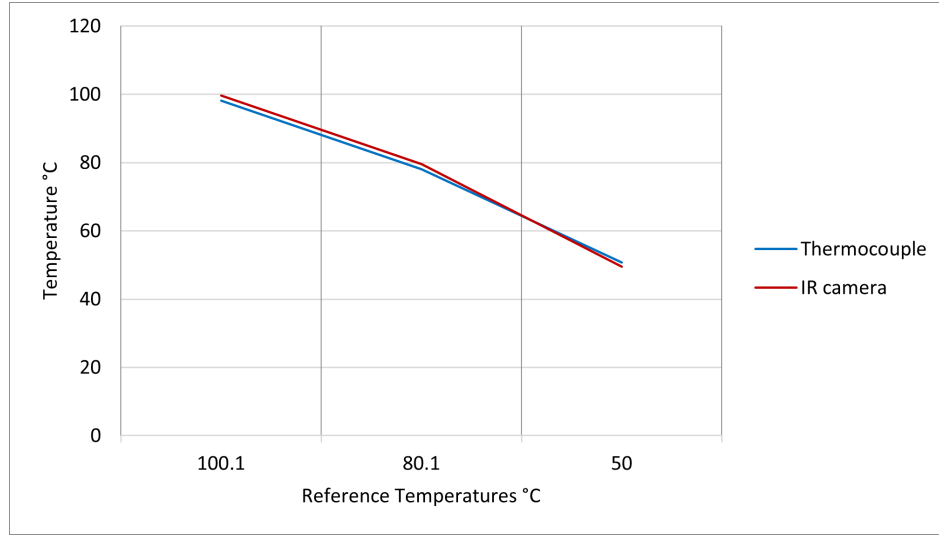
Data logger channels are limited, so recording the temperature of every point on the product won't be possible. To overcome this problem, IR cameras are used to

determine the heat spots on the product and the surface temperature distributions of the critical regions of the system. The tailored cooling system mainly involves forced convection heat transfer and airflow through a fan channel. IR camera allows us to determine the changes in the temperature variations in the observed region [31]. Figure 15 shows the IR camera used for this study. Before the temperature measurement, IR camera calibration is necessary. To conduct calibration, it is necessary to employ thermal imaging to assess the temperature of an object with a known thermal level. This procedure must be performed in a controlled setting isolated from ambient radiation. Blackbody objects, characterized by their ability to emit and absorb nearly all forms of electromagnetic radiation, are implemented in this process. This step is critical as these blackbody objects are subjected to specific temperatures during the calibration process, and they must emit almost all of the thermal energy generated by their heating so that the emitted infrared radiation is coherent with the temperature [32].



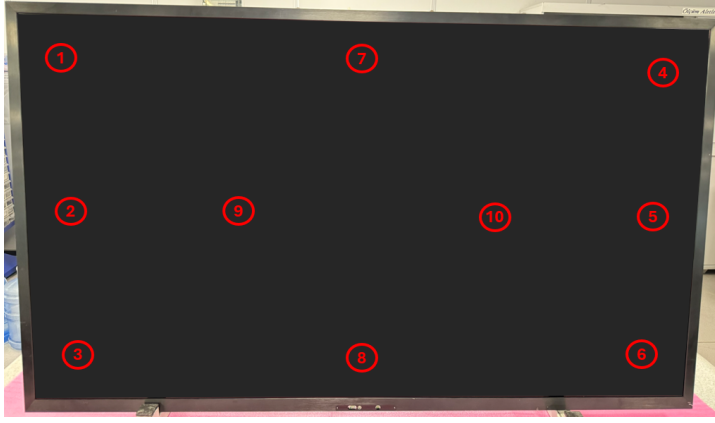
**Figure 15:** FLIR E60 brand IR camera for temperature measurements.

After calibrating the IR camera, the thermocouple and IR camera measurements were compared using reference temperatures from calibrator surfaces. The IR camera readings are close to the thermocouple readings. From Figure 16, the percentage deviation between the thermocouple and IR camera measurements increases to a maximum of 2.4%.

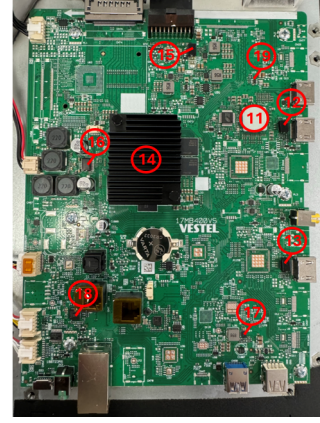


**Figure 16:** Temperature measurement comparison for IR camera and thermocouple with respect to reference temperatures.

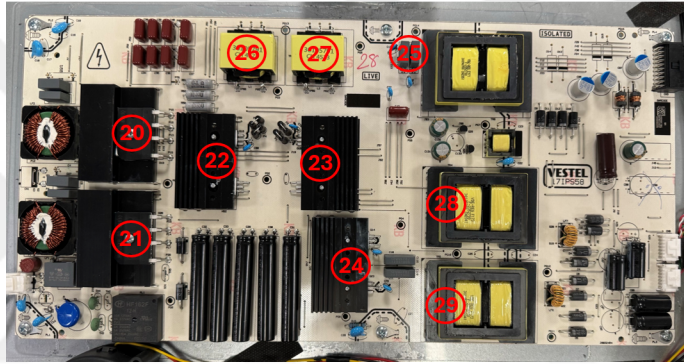
Thermocouples are attached to critical components, especially on MB and PB. Cell temperatures are also critical, so thermocouples attach to the cell surface to control the temperature variations on the cell surface. A stationary thermal camera is adjusted at the back side of the product to control the temperature distribution of the flow through the fan channel. Figure 17 shows the predetermined measurement points for thermocouples on cell, MB, and PB.



(a)



(b)



(c)

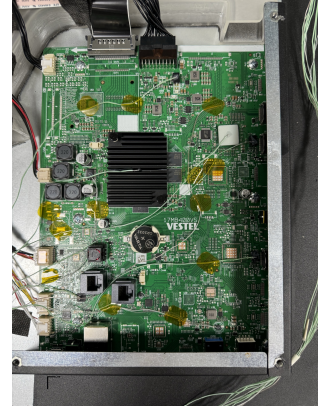
**Figure 17:** Thermocouple attachment points for the cell (a), MB (b), and PB (c).

After determining the measurement points, thermocouples are attached to the components for the experiment. Before attaching the thermocouples, the application surface must be cleaned with isopropyl alcohol (IPA). Following the cleaning process, cyanoacrylate adhesive is applied to bond the thermocouple securely onto the prepared surface. Figure 18 shows the experimental setup and thermocouple attachments on the product.

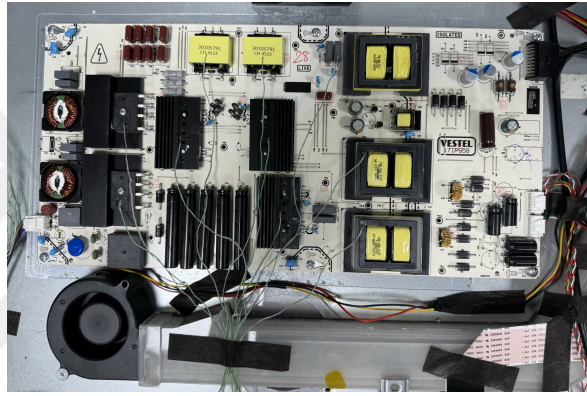




(a)



(b)



(c)

**Figure 18:** Thermocouple attachments of the cell (a), MB (b), and PB (c) for the experimental setup.

The device was subjected to environmental testing in a controlled heat room, maintaining a temperature of  $40^{\circ}\text{C}$  and 90% relative humidity. The temperature inside the heat-controlled room is controlled by the on/off controller. During this period, the product is loaded in maximum operating conditions (sound and brightness are at maximum level, and all features are active). The time it takes for the product to reach a steady state may vary from product to product. For display devices, it is accepted as an average of 4 hours. Theoretically, stability can be accepted as the difference between the component surface temperatures does not exceed  $3^{\circ}\text{C}$  in 1 minute and is checked by the multichannel data logger.

Figure 19 shows the product inside the heat room subjected to testing.



**Figure 19:** Image of the product placed in the heat room for testing.

## CHAPTER IV

### RESULTS

#### *4.1 Analytical Results*

For the product with a heuristic cooling system, the operating velocities of the fans can be calculated using a couple of assumptions explained in Chapter 2.

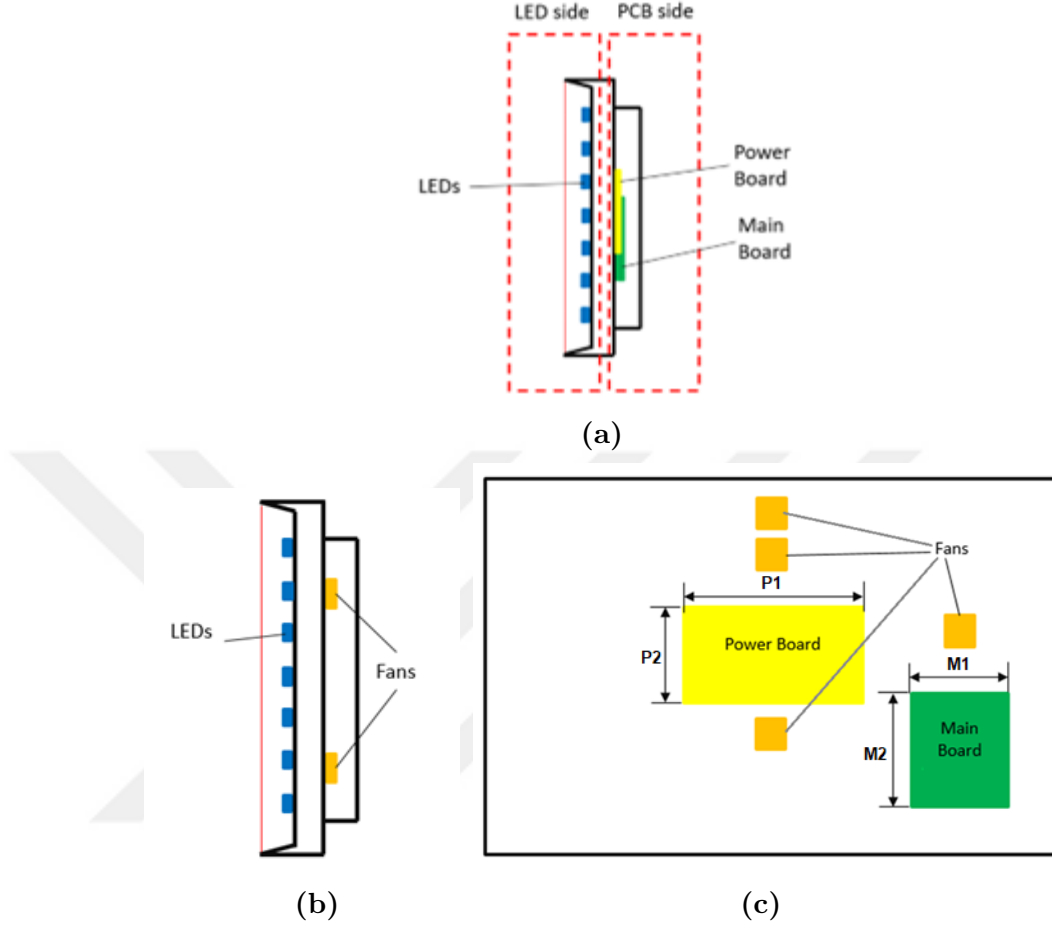
The product with a heuristic cooling system is modeled with the assumption of external flow over vertical and horizontal plates. Using Equation 19 and Equation 20, the required natural convection heat transfer coefficient for the surface temperature of the components can be found.

MB and PB components are assumed to operate properly at around 100 °C surface temperature, considering component specifications and the derating rule for reliability. Using Equation 16, the film temperature was calculated as 70 °C. All fluid properties were obtained using 70°C film temperature for analytical solutions from the air properties tables [3].

Analytical solutions are performed on the product with the heuristic cooling system before the numerical simulations to determine its cooling performance.

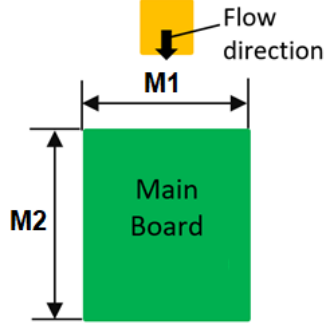
First, fans of the heuristic cooling system are examined to determine whether they are enough to dissipate the heat generated. The current system is divided into two sections: the LED side and the PCB side (Figure 20).



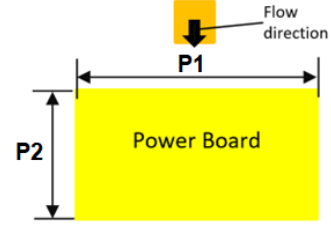


**Figure 20:** The current system divided into sections (a). LED section (b) and PCB section (c) of the current system for analytical study.

$P1$  and  $P2$  represent the dimensions of the PB.  $M1$  and  $M2$  represent the dimensions of MB. The analytical study first focuses on the PCB side of the heuristic cooling system.



**Figure 21:** Dimensions and fan flow direction of MB.



**Figure 22:** Dimensions and fan flow direction of PB.

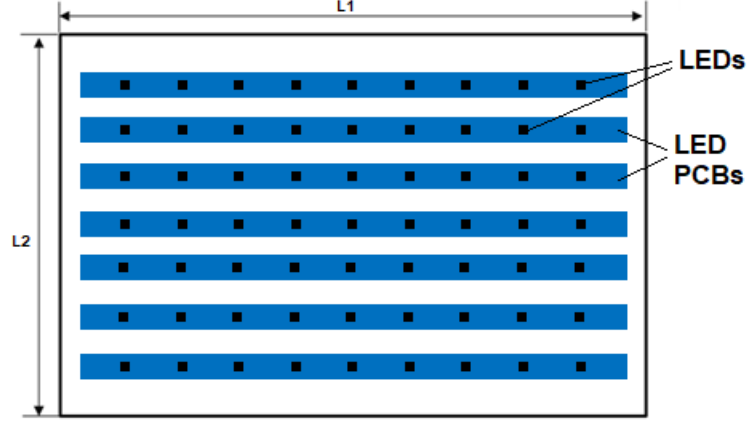
Using Equation 18, the required heat transfer coefficient for maintaining the surface temperatures set for the MB and PB can be found. After finding the heat transfer coefficient, Equation 14, Equation 19, and Equation 20 are used to determine fan velocities to maintain the predetermined surface temperature of the mainboard, powerboard, and its components. After a set of calculations, the fan speeds needed to maintain the predetermined surface temperature of MB and PB are 6.3 m/s and 1.0 m/s, respectively.

Table 2 shows the radial fan properties of the current cooling system to cool the MB and PB, which were created by heuristic approaches.

**Table 2:** Radial fan properties.

| Fan Name   | Power (W) | Speed (rpm) | Air Flow (CFM) | Velocity (m/s) | Noise (dB) | Fan Opening |      |
|------------|-----------|-------------|----------------|----------------|------------|-------------|------|
|            |           |             |                |                |            | x(m)        | y(m) |
| Radial Fan | 2.4       | 3400        | 15             | 6.3            | 42         | 0.03        | 0.04 |

The same calculations are conducted for the LED side of the system as on the PCB side. The fan speed required to maintain the predetermined surface temperature of the LED side is calculated as 0.5 m/s. L1 and L2 represent the outer dimension of the product.



**Figure 23:** Dimension information of LED side of the system and components.

Table 3 shows the axial fan properties of the current cooling system for cooling the LED side of the product, which were created using heuristic approaches.

**Table 3:** Axial fan properties

| Fan Name  | Power (W) | Speed (rpm) | Air Flow (CFM) | Velocity (m/s) | Noise (dB) | Fan Opening |      |
|-----------|-----------|-------------|----------------|----------------|------------|-------------|------|
|           |           |             |                |                |            | x(m)        | y(m) |
| Axial Fan | 0.8       | 2400        | 29.6           | 4.3            | 25         | 0.07        | 0.03 |

Table 4 compares the fan speeds in the specifications and those achieved by the analytical approach.

**Table 4:** Comparison of the analytical fan speeds and the speeds in the specifications of the heuristic cooling system.

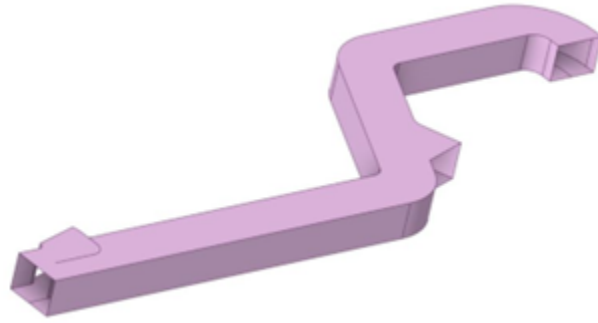
|                       | Specifications | Analytical Results |
|-----------------------|----------------|--------------------|
| Radial Fan Speed (MB) | 6.3 m/s        | 6.3 m/s            |
| Radial Fan Speed (PB) | 6.3 m/s        | 1.0 m/s            |
| Axial Fan Speed       | 4.3 m/s        | 0.5 m/s            |

From Table 2, the maximum fan speed is 6.3 m/s. According to analytical calculations, the radial fan speed must be at least 6.3 m/s from Table 4 to maintain the desired surface temperature of MB. The current fan speed is enough to cool the MB. Unlike MB, the fan speed must be 1.0 m/s minimum to maintain the desired surface

temperature of PB. The current fan speed of 6.3 m/s is nearly six times greater than the requirement. This meets the fan speed requirement. However, it means that over-spec fan usage. From Table 3, the maximum fan speed is 4.3 m/s. The fan speed must be at least 0.5 m/s from Table 4 to maintain the desired surface temperature of the LED side of the product. The current fan speed is enough to cool the LED side of the system. However, it means that over-spec fan usage.

For this reason, the cooling design of the system can be optimized by reducing the cooling system elements.

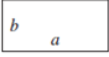
An analytical study of the fan channel is carried out for the product using a tailored cooling system. Figure 24 shows the fan channel system selected for the prototyping.



**Figure 24:** Radial fan channel.

Using this radial fan channel structure, the system curve must be determined by calculating pressure drops through the fan channel for different low rates of the fan. Subsequently, the system and fan curves are compared to identify the optimum operating conditions for the radial fan.

First, the hydraulic diameter is found by using Equation 21. Equation 22 is used to determine the effective diameter of the system. The effective diameter is then calculated using Figure 25, which provides the laminar loss coefficient.

| Rectangular                                                                       |              |
|-----------------------------------------------------------------------------------|--------------|
|  |              |
| $b/a$                                                                             | $f Re_{D_h}$ |
| 0.0                                                                               | 96.00        |
| 0.05                                                                              | 89.91        |
| 0.1                                                                               | 84.68        |
| 0.125                                                                             | 82.34        |
| 0.167                                                                             | 78.81        |
| 0.25                                                                              | 72.93        |
| 0.4                                                                               | 65.47        |
| 0.5                                                                               | 62.19        |
| 0.75                                                                              | 57.89        |
| 1.0                                                                               | 56.91        |

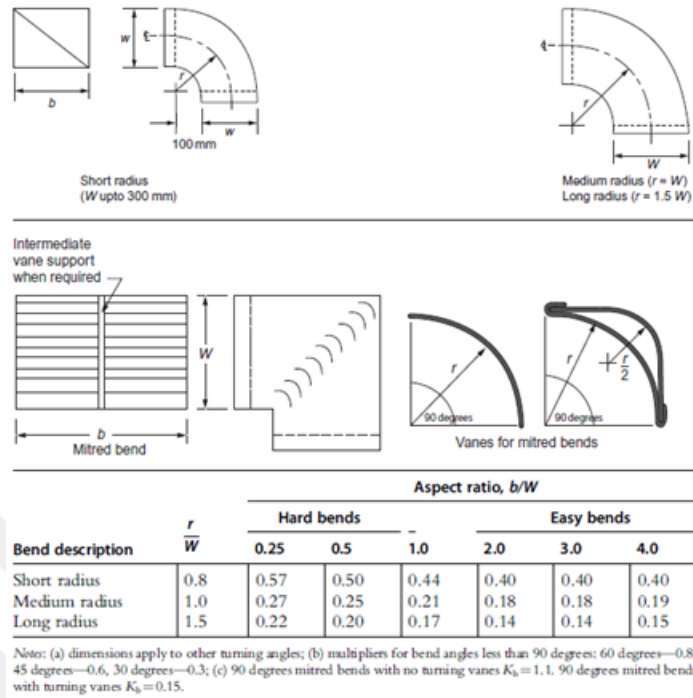
**Figure 25:** Laminar friction constant of rectangular duct [3].

The duct material will be plastic. From Figure 26, the surface roughness of the plastic can be found.

| $\epsilon$ |                  |          |        |                |
|------------|------------------|----------|--------|----------------|
| Material   | Condition        | ft       | mm     | Uncertainty, % |
| Steel      | Sheet metal, new | 0.00016  | 0.05   | $\pm 60$       |
|            | Stainless, new   | 0.000007 | 0.002  | $\pm 50$       |
|            | Commercial, new  | 0.00015  | 0.046  | $\pm 30$       |
|            | Riveted          | 0.01     | 3.0    | $\pm 70$       |
|            | Rusted           | 0.007    | 2.0    | $\pm 50$       |
| Iron       | Cast, new        | 0.00085  | 0.26   | $\pm 50$       |
|            | Wrought, new     | 0.00015  | 0.046  | $\pm 20$       |
|            | Galvanized, new  | 0.0005   | 0.15   | $\pm 40$       |
|            | Asphalted cast   | 0.0004   | 0.12   | $\pm 50$       |
| Brass      | Drawn, new       | 0.000007 | 0.002  | $\pm 50$       |
| Plastic    | Drawn tubing     | 0.000005 | 0.0015 | $\pm 60$       |
| Glass      | —                | Smooth   | Smooth |                |
| Concrete   | Smoothed         | 0.00013  | 0.04   | $\pm 60$       |
|            | Rough            | 0.007    | 2.0    | $\pm 50$       |
| Rubber     | Smoothed         | 0.000033 | 0.01   | $\pm 60$       |
| Wood       | Stave            | 0.0016   | 0.5    | $\pm 40$       |

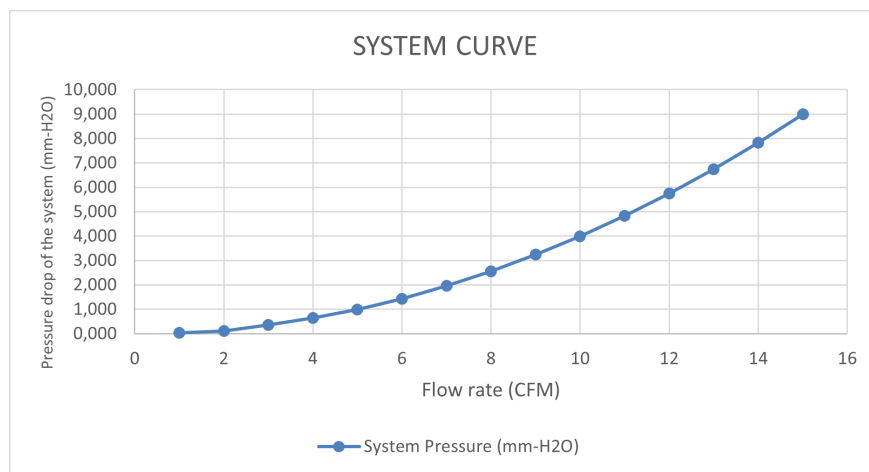
**Figure 26:** Recommended roughness values of commercial ducts [3].

To determine the loss of the system, the friction factor must be found using Moody's Chart. The Reynolds number and relative roughness are required to determine the friction factor of the system. Using Equation 23, relative roughness can be found. The total pressure drop of the system can be calculated by combining friction losses and minor losses using Equation 24.  $K$  is the minor loss coefficient. Our system has one pair of  $90^\circ$  duct bends and two pairs of  $60^\circ$  duct bends. According to Figure 27, the loss coefficients of our  $90^\circ$  duct bends and  $60^\circ$  bends can be easily calculated.

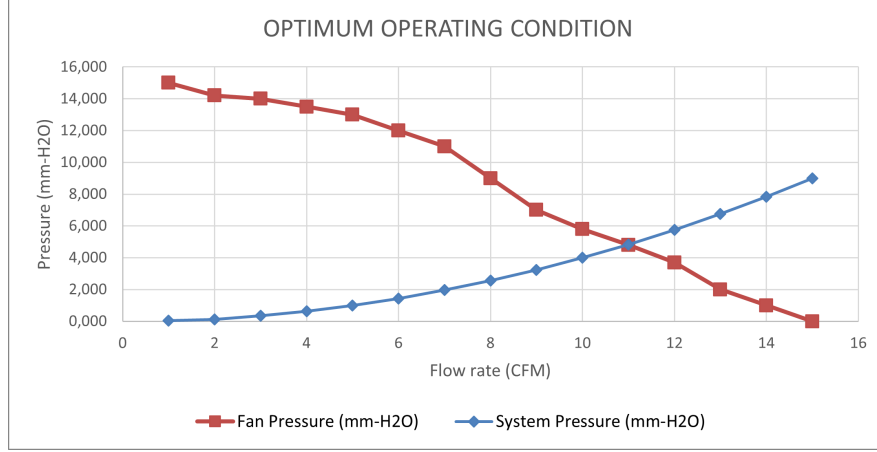


**Figure 27:** Loss coefficient of rectangular duct bends [4].

Figure 28 and Figure 29 represent the system curve and the combination of the system and fan curves, respectively.



**Figure 28:** Pressure drop of the system.



**Figure 29:** Combined curve that indicates optimum operating flow rate of the system.

According to Figure 29, the fan's optimum operating flow rate is around 11 cfm, and the pressure is around 4.83mm-H2O.

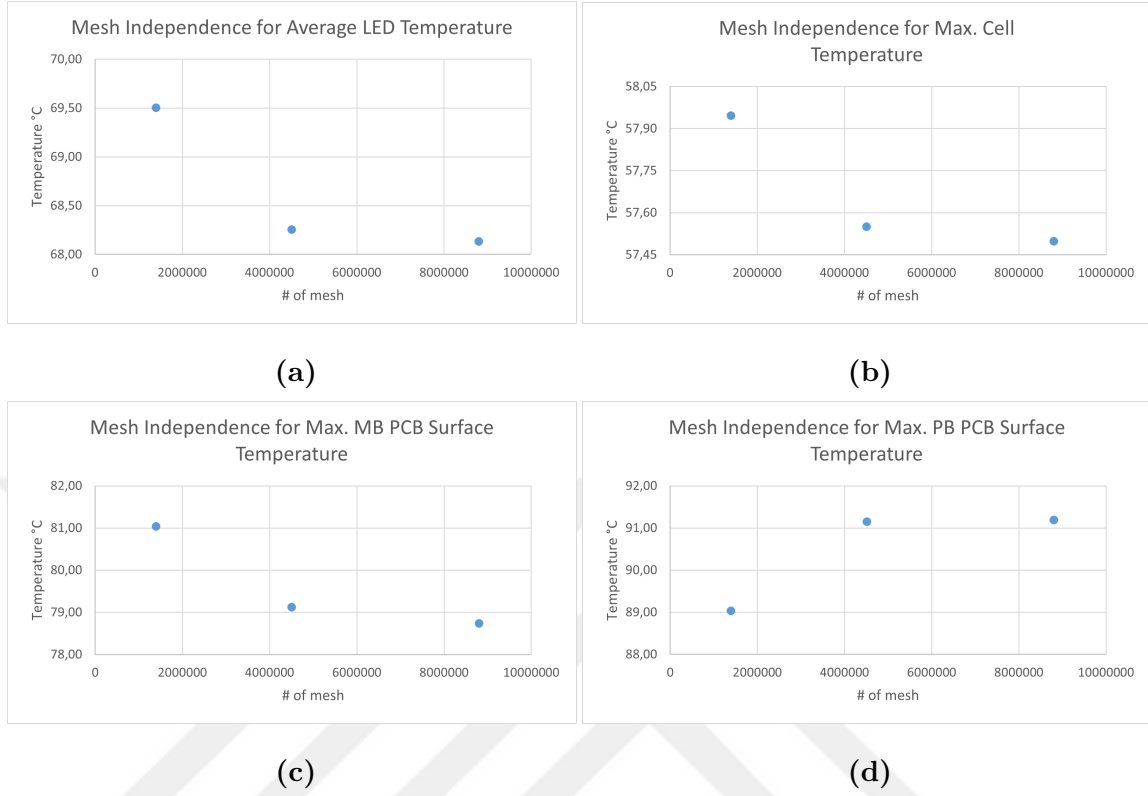
## 4.2 Numerical Results

The products with heuristic and tailored cooling systems are analyzed numerically. The effect of reducing the number of fans on the cooling performance of the product with a tailored cooling system is examined. LED, Cell, PB, and MB temperatures are recorded for further comparison.

### 4.2.1 Mesh Sensitivity Study

Mesh independence is an important step in simulations. The given values and information from the simulations must be mesh-independent. In this study, all results are mesh-independent. The temperature comparison tables for MB and PB were achieved after comparing the results with different numbers of meshes.

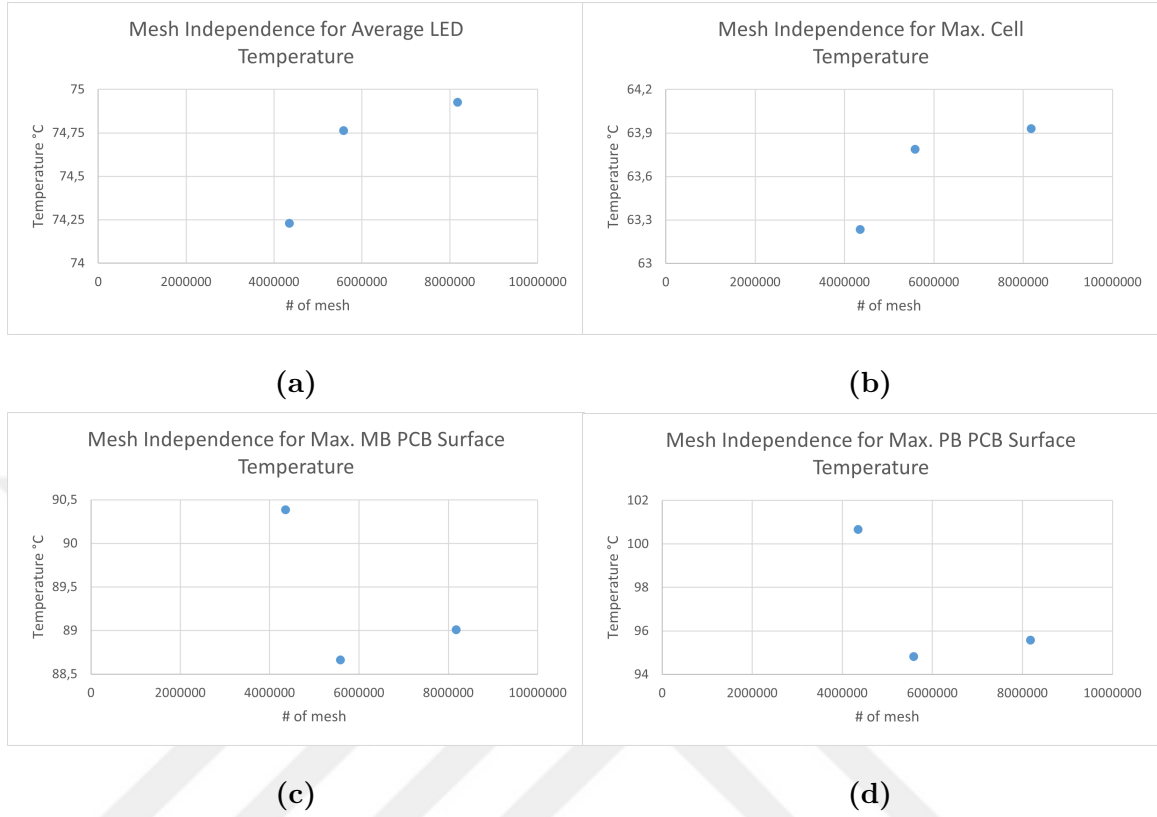
Figure 30 shows the mesh-independent solutions for the max surface temperatures of MB, PB, and cell. It also represents the mesh independence of the average LED temperature for the heuristic cooling system.



**Figure 30:** Mesh sensitivity study of the product with the heuristic cooling system. (a) Mesh independence of average LED temperature. (b) Mesh Independence of max. cell temperature. (c) Mesh Independence of max. MB PCB surface temperature. (d) Mesh Independence of max. PB PCB surface temperature.

Figure 31 shows the mesh-independent solutions for the max surface temperatures of MB, PB, and cell. It also represents the mesh independence of the average LED junction temperature for the tailored cooling system.



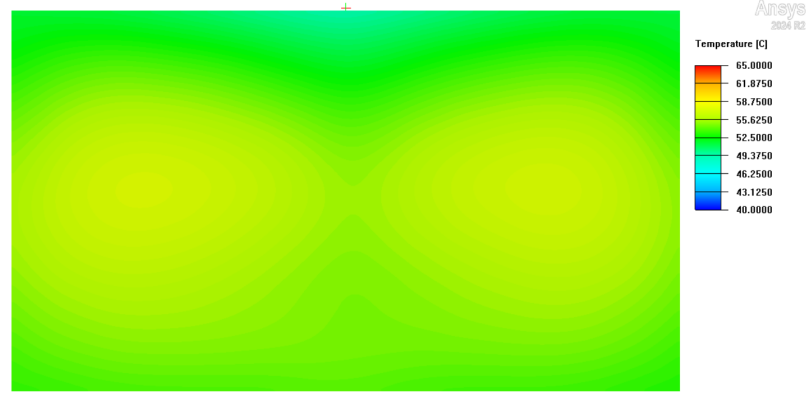


**Figure 31:** Mesh sensitivity study of the product with tailored cooling system. (a) Mesh independence of average LED temperature. (b) Mesh Independence of max. cell temperature. (c) Mesh Independence of max. MB PCB surface temperature. (d) Mesh Independence of max. PB PCB surface temperature.

The examples of mesh independence can be increased for the specific components and parts within the heuristic and tailored cooling systems.

#### 4.2.2 Numerical Results of The Product With Heuristic Cooling System

Simulations with ANSYS Icepak show the system's temperature distribution. Figure 32 shows the temperature distribution of the cell that was achieved with the heuristic approach.



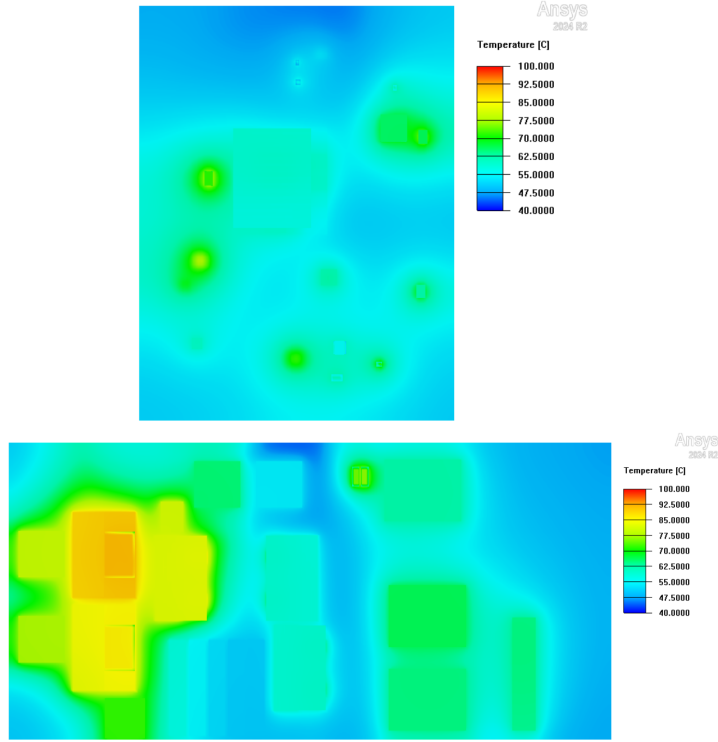
**Figure 32:** Numerical simulation results that show cell temperatures from different locations.

Table 5 shows the simulation results for the cell temperatures of the heuristic cooling system.

**Table 5:** Numerical measurements on the cell surface of the heuristic cooling system.

| MEASUREMENT POINTS | CELL TEMPERATURES (°C) |
|--------------------|------------------------|
| 1                  | 53.0                   |
| 2                  | 56.6                   |
| 3                  | 54.5                   |
| 4                  | 52.7                   |
| 5                  | 56.4                   |
| 6                  | 54.3                   |
| 7                  | 51.2                   |
| 8                  | 54.7                   |
| 9                  | 57.1                   |
| 10                 | 57.2                   |

Figure 33 and Table 6 shows the temperature distribution of MB and PB components with the heuristic cooling system.



**Figure 33:** Numerical results for temperature distribution of the MB and PB components with the heuristic cooling system.

**Table 6:** Numerical measurement on the MB and PB component surfaces of the heuristic cooling system.

| MEASUREMENT POINTS | MB TEMPERATURES (°C) | MEASUREMENT POINTS | PB TEMPERATURES (°C) |
|--------------------|----------------------|--------------------|----------------------|
| 11                 | 65.7                 | 20                 | 91.6                 |
| 12                 | 66.2                 | 21                 | 86.2                 |
| 13                 | 62.7                 | 22                 | 82.5                 |
| 14                 | 59.6                 | 23                 | 59.1                 |
| 15                 | 54.0                 | 24                 | 59.6                 |
| 16                 | 71.5                 | 25                 | 76.0                 |
| 17                 | 65.1                 | 26                 | 65.1                 |
| 18                 | 60.4                 | 27                 | 53.7                 |
| 19                 | 58.3                 | 28                 | 66.4                 |
|                    |                      | 29                 | 64.5                 |

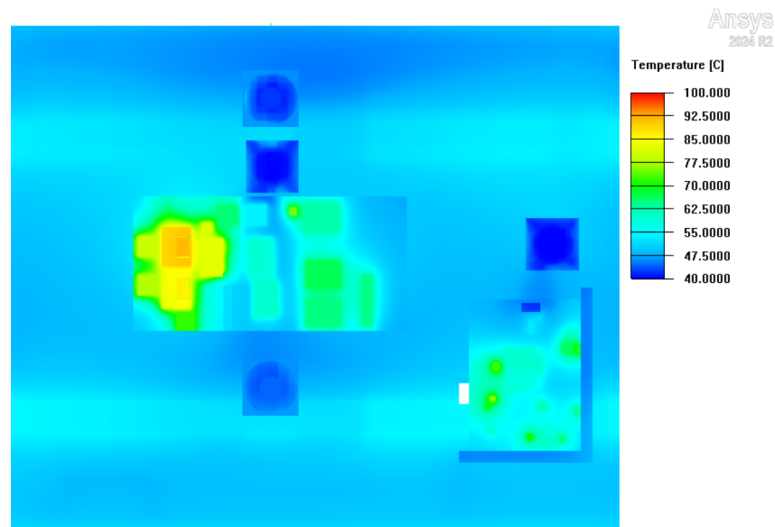
Figure 34 shows the temperature distribution of LED PCBs and LEDs of the

product with the heuristic cooling system.



**Figure 34:** Numerical results for temperature distribution of LED PCBs and LEDs of the product with heuristic cooling system.

Figure 35 shows the temperature distribution of the back side of the product with the heuristic cooling system to create an understanding of the overall effect of the cooling components on the product.

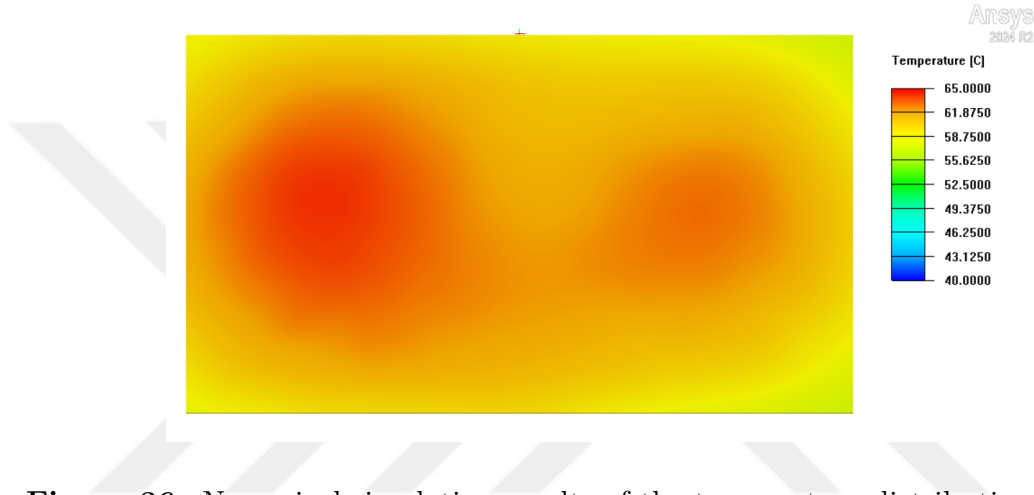


**Figure 35:** Numerical results for temperature distribution of the back side of the product with heuristic cooling system.

### 4.2.3 Numerical Results of The Product With Tailored Cooling System

For the entire product, the following figures and tables represent the temperature distributions.

Figure 36 and Table 7 show the temperature distribution on the cells with the tailored cooling system.

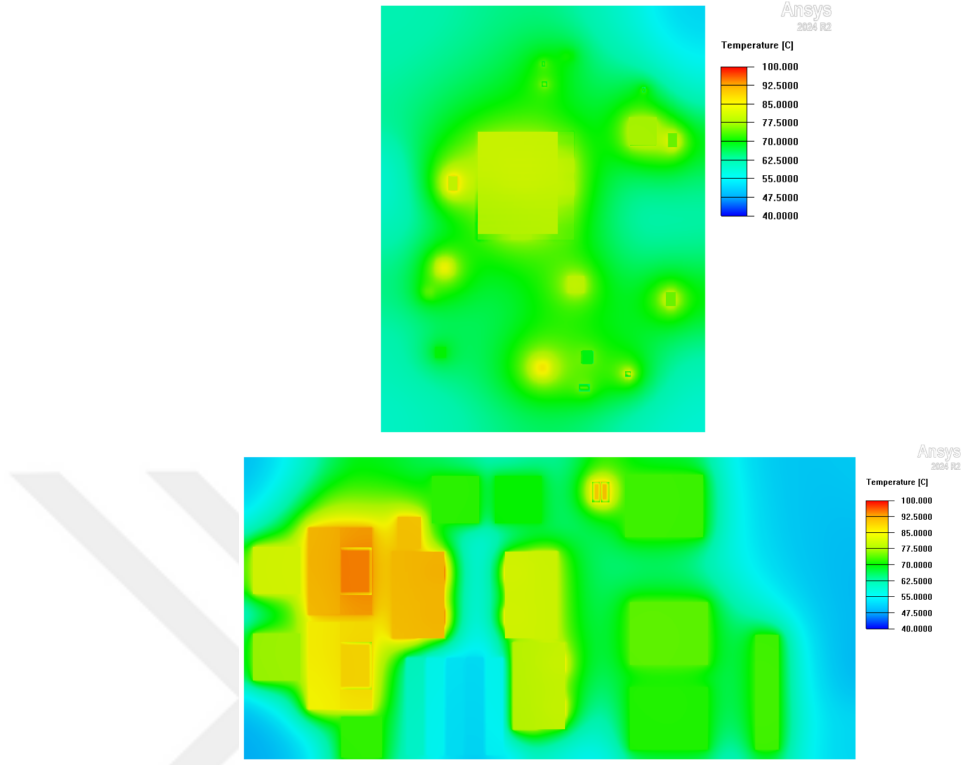


**Figure 36:** Numerical simulation results of the temperature distribution of the cell with tailored cooling system.

**Table 7:** Numerical measurement on the cell surface of the tailored cooling system.

| MEASUREMENT POINTS | CELL TEMPERATURES (°C) |
|--------------------|------------------------|
| 1                  | 60.5                   |
| 2                  | 63.1                   |
| 3                  | 60.9                   |
| 4                  | 59.5                   |
| 5                  | 61.9                   |
| 6                  | 59.5                   |
| 7                  | 60.4                   |
| 8                  | 61.2                   |
| 9                  | 63.9                   |
| 10                 | 62.5                   |

Figure 37 and Table 8 shows the temperature distribution of MB and PB components with the tailored cooling system.



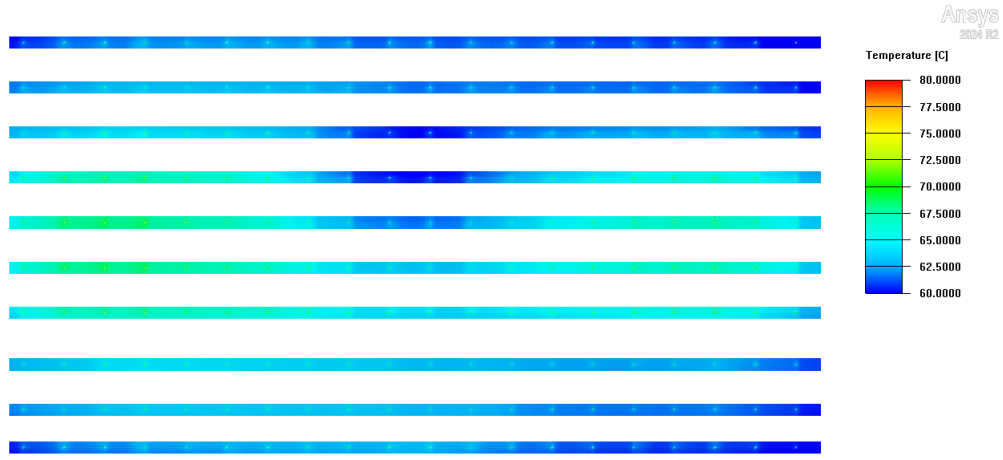
**Figure 37:** Numerical results for temperature distribution of the MB and PB components with tailored cooling system.

**Table 8:** Numerical measurement on the MB and PB component surfaces of the tailored cooling system.

| MEASUREMENT POINTS | MB TEMPERATURES (°C) | MEASUREMENT POINTS | PB TEMPERATURES (°C) |
|--------------------|----------------------|--------------------|----------------------|
| 11                 | 77.4                 | 20                 | 94.6                 |
| 12                 | 76.7                 | 21                 | 88.6                 |
| 13                 | 74.7                 | 22                 | 91.5                 |
| 14                 | 80.6                 | 23                 | 81.3                 |
| 15                 | 73.2                 | 24                 | 79.8                 |
| 16                 | 79.8                 | 25                 | 90.4                 |
| 17                 | 80.8                 | 26                 | 71.9                 |
| 18                 | 69.9                 | 27                 | 70.3                 |
| 19                 | 70.9                 | 28                 | 75.2                 |
|                    |                      | 29                 | 74.3                 |

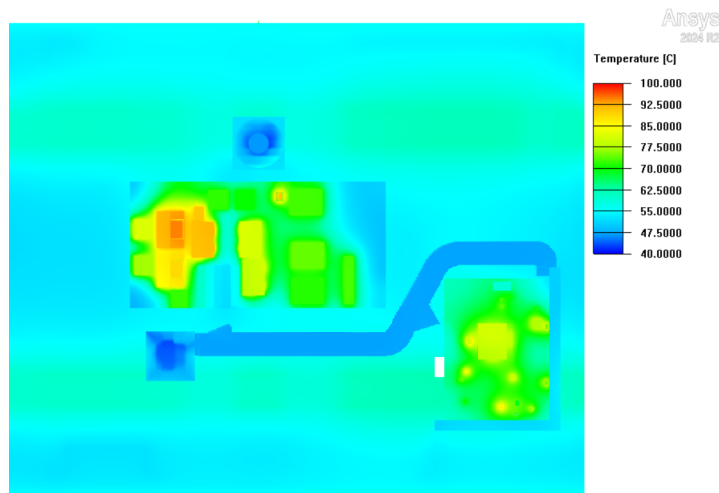
Figure 38 shows the temperature distribution of LED PCBs and LEDs of the

product with the tailored cooling system.



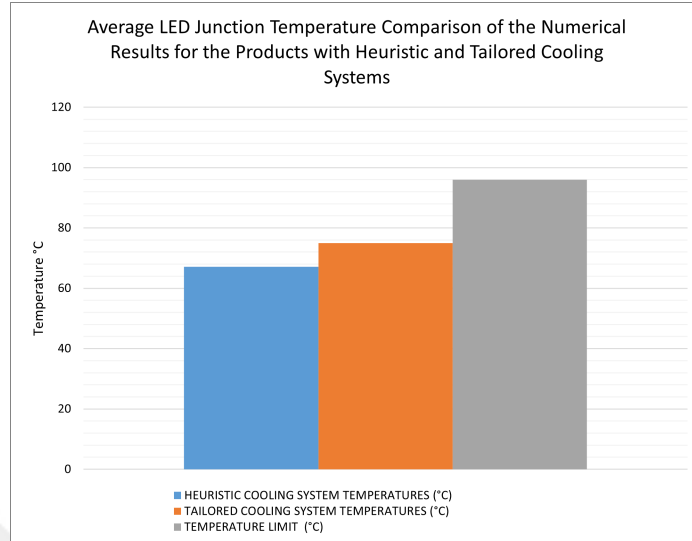
**Figure 38:** Numerical results for temperature distribution of LED PCBs and LEDs of the product with tailored cooling system.

Figure 39 shows the temperature distribution of the back side of the product with the tailored cooling system.



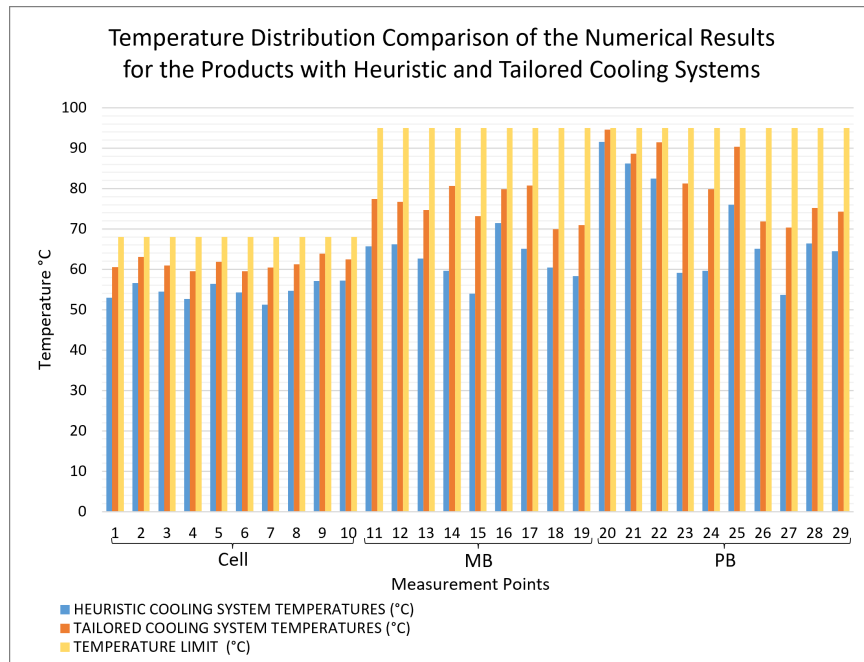
**Figure 39:** Numerical results for the temperature distribution of the back side of the product with tailored cooling system.

Figure 40 compares the LED junction numerical temperature measurement results of both heuristic and tailored cooling systems.



**Figure 40:** Comparison of the numerical results for the average LED junction temperature of the product with tailored cooling system.

Figure 41 compares numerical temperature measurement results of both heuristic and tailored cooling systems.



**Figure 41:** Temperature distribution comparison of the numerical results for the products with heuristic and tailored cooling systems.

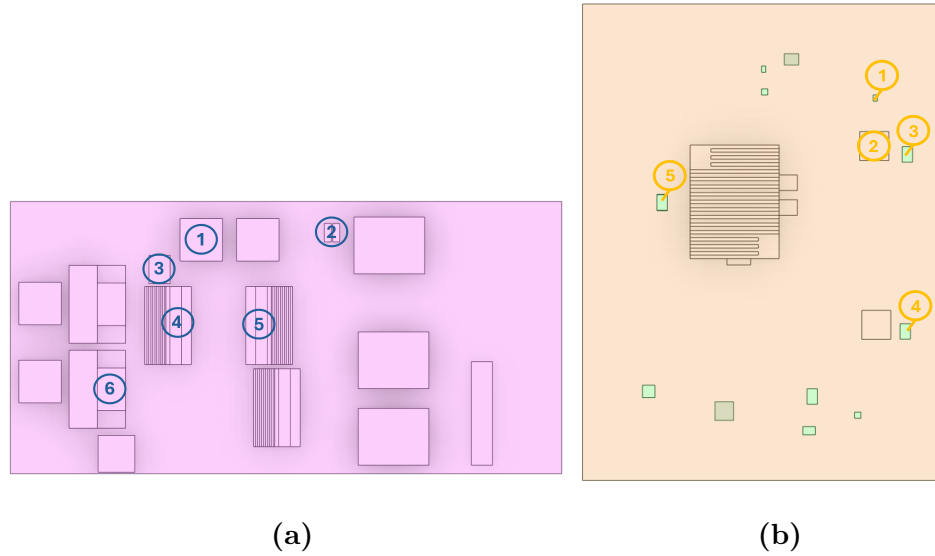


In Figure 40 and Figure 41, it is obvious that the temperature results from the simulations of the tailored cooling system are higher than those of the heuristic cooling system. Some of the components on MB and PB are close to the temperature limits. However, none of the LEDs, cells, MB, and PB exceed the temperature limits for the tailored cooling system. Theoretically, moving with a tailored cooling system for the experiments is acceptable.

#### 4.2.3.1 Selection of Tailored Design From Candidates

Fan channel is an important component of the tailored cooling system, and it is created with the help of simulations among the candidate designs, as explained in Chapter 2.

The best option among the designs represented in Figure 9 is selected by comparing the spot temperatures of the same points on MB and PB by numerical simulations. Figure 42 represents the temperature measurement points from simulations for comparisons.



**Figure 42:** Temperature measurement points from MB (a) and PB (b) for temperature comparisons of candidate fan channel designs.

Table 9 and Table 10 show the temperature differences between the same points on the MB and PB after numerical simulations.

**Table 9:** Temperature comparison of MB components for different channel structures.

| MB TEMPERATURES (°C) |          |      |      |      |      |
|----------------------|----------|------|------|------|------|
| Measurement Points   | (a)      | (b)  | (c)  | (d)  | (e)  |
| 1                    | 87.7     | 87.4 | 85.8 | 80.7 | 78.8 |
| 2                    | 82.3     | 89.7 | 85.1 | 85.1 | 86.3 |
| 3                    | 82.3     | 87.1 | 82.2 | 81.6 | 82.8 |
| 4                    | 94.1     | 96.3 | 91.9 | 90.9 | 87.3 |
| 5                    | 94.8     | 94.5 | 99.9 | 87.9 | 85.4 |
|                      | AVERAGES |      |      |      |      |
|                      | 88.2     | 91.0 | 89.0 | 85.2 | 84.1 |

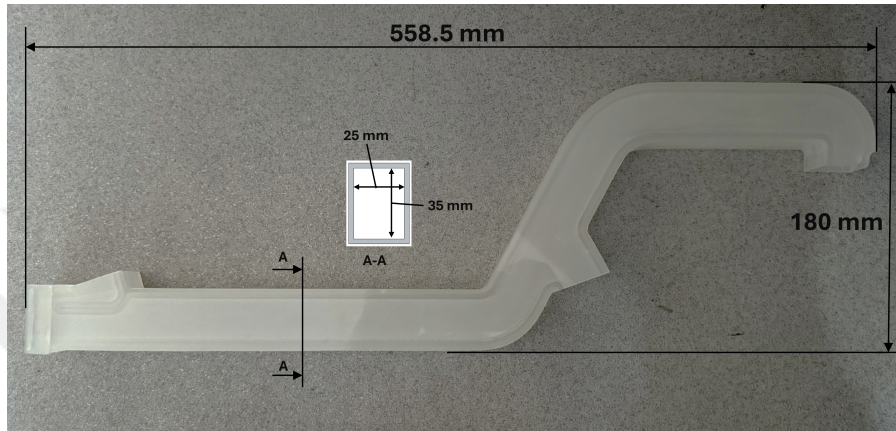
**Table 10:** Temperature comparison of PB components for different channel structures.

| MB TEMPERATURES (°C) |          |      |      |      |      |
|----------------------|----------|------|------|------|------|
| Measurement Points   | (a)      | (b)  | (c)  | (d)  | (e)  |
| 1                    | 84.5     | 80.9 | 80.7 | 70.4 | 69.9 |
| 2                    | 67.0     | 81.2 | 74.4 | 82.2 | 85.1 |
| 3                    | 93.6     | 89.9 | 90.7 | 71.9 | 80.8 |
| 4                    | 88.7     | 88.7 | 88.4 | 87.3 | 79.1 |
| 5                    | 92.1     | 74.2 | 77.7 | 78.7 | 93.5 |
| 6                    | 76.9     | 78.4 | 81.1 | 71.6 | 84.2 |
|                      | AVERAGES |      |      |      |      |
|                      | 83.8     | 82.2 | 82.2 | 77.0 | 82.1 |

Based on the average temperatures in the tables, design (e) in Table 9 is the best for MB cooling; however, design (d) in Table 10 is the best for PB cooling. At this point, the overall reliability and life of the product must be considered. The lower the temperatures on the MB, the longer the life of the processors and other critical

components for the display device's functions. For the experimental work, design (e) is chosen for prototyping.

Figure 43 shows the fan channel prototype before it is assembled into the product with the tailored cooling system.



**Figure 43:** The prototype of the selected fan channel design and its critical dimensions.

#### 4.2.3.2 Comparison of Numerical and Analytical Fan Operating Points

The analytical and numerical solutions of the tailored cooling system are compared in terms of fan operating points.

The fan operating points from the simulation are presented in Figure 44 and Table 11.

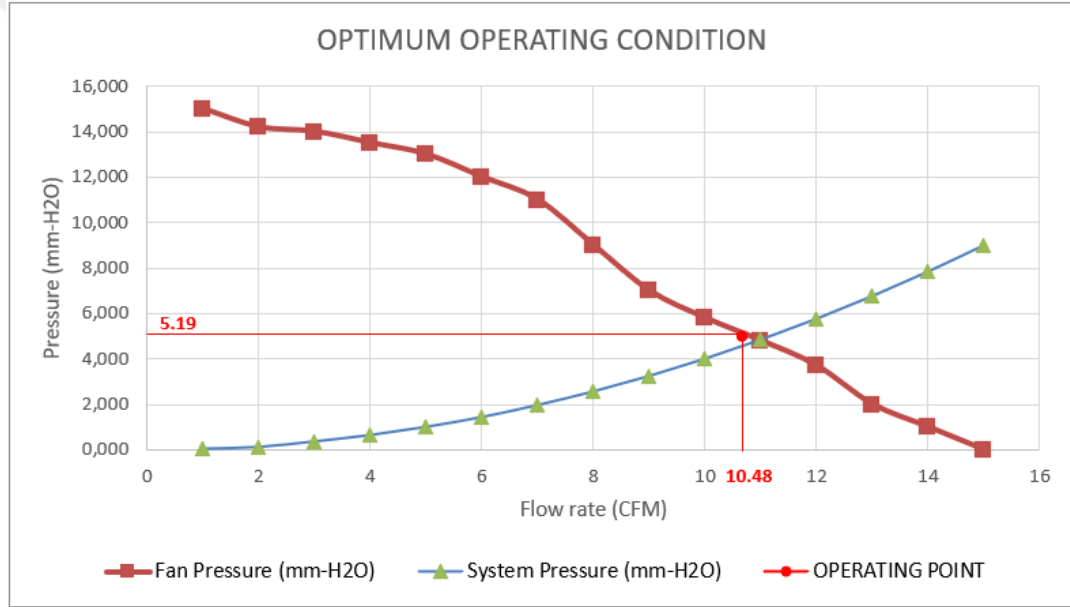
| Fan operating points                                                                 |                             |                            |
|--------------------------------------------------------------------------------------|-----------------------------|----------------------------|
| Object names                                                                         | Volume flow                 | Pressure rise              |
| Sunon.PMB2475PNB2-AY_2.1                                                             | 4.944e-03 m <sup>3</sup> /s | 50.873742 N/m <sup>2</sup> |
| <div> <input type="button" value="Ok"/> <input type="button" value="Export"/> </div> |                             |                            |

**Figure 44:** Operating point of the radial fan for the optimized system according to simulation results.

**Table 11:** Operating point of the system according to simulation results.

| Fan Operating Conditions |         |        |
|--------------------------|---------|--------|
| Flow Rate                | $m^3/s$ | cfm    |
|                          | 0.0049  | 10.48  |
| Pressure drop            | Pa      | mm-H2O |
|                          | 50.87   | 5.19   |

Figure 45 shows the comparison of the radial fan's analytical and numerical operating conditions,



**Figure 45:** Comparison of the operating point of the radial fan for the optimized system according to simulation results and analytical results.

Table 12 shows the percentage of the difference between analytical and numerical solutions for radial fan operating conditions.

**Table 12:** Analytical and Numerical solution comparison of radial fan operating conditions.

| Fan Operating Conditions               | Analytical Solution | Numerical Solution | % Difference |
|----------------------------------------|---------------------|--------------------|--------------|
| Flow rate<br>(cfm)                     | 11                  | 10.48              | 4.96         |
| Pressure drop<br>(mm-H <sub>2</sub> O) | 4.83                | 5.19               | -6.94        |

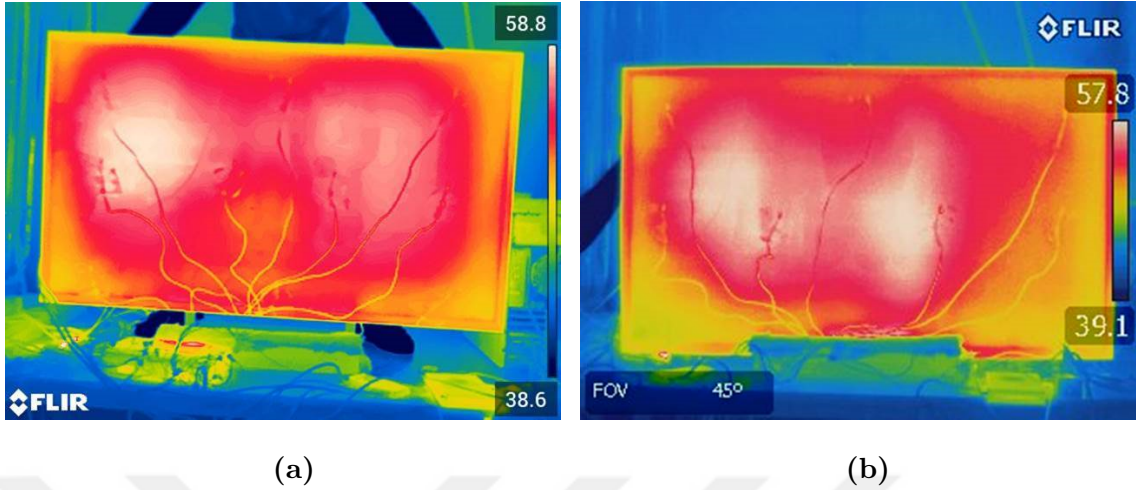
### 4.3 *Experimental Results and Comparison*

To ensure accurate data collection from the system, it was allowed to reach a steady state before obtaining temperature readings and thermal camera images. The products were tested in the heat room. During this period, temperature readings with thermocouples and thermal camera images were collected from the device to understand the cooling system's performance under 40 °C and 90% relative humidity conditions.

#### 4.3.1 **Experimental Results of the Product With Heuristic and Tailored Cooling Systems**

The products with a heuristic and tailored cooling system were tested in the heat room, and the results are represented and compared in the following images and tables.

Figure 46 and Table 13 show the temperature distribution of the cell with the heuristic and tailored cooling system.



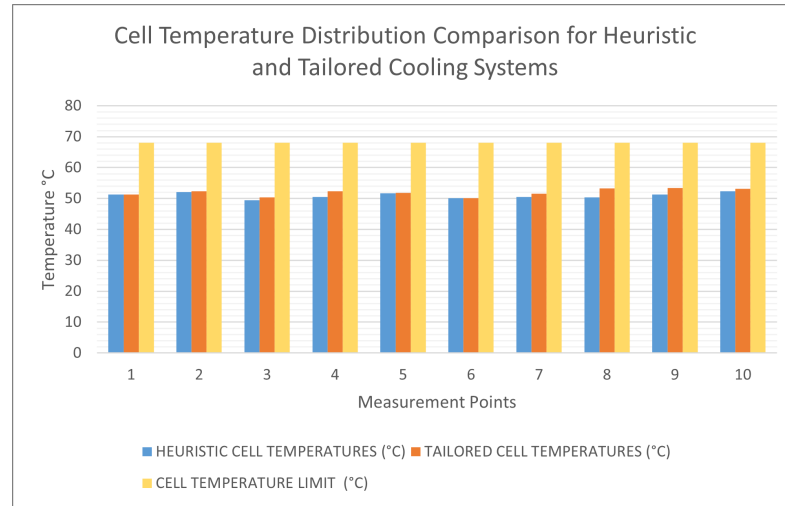
**Figure 46:** Thermal image of the temperature distribution of the cell. (a) Heuristic cooling system. (b) Tailored cooling system.

From Figure 46, temperature distribution on cells for heuristic and tailored cooling systems is very similar. Reducing one axial and one radial fan has no significant effect on the cell temperature in the tailored cooling system. The cell is cooled down in the product with a tailored cooling system with only one axial fan. In the product with the heuristic cooling system, the cell is cooled down with two axial fans. The temperature distribution of the cells is similar in both cases.

**Table 13:** The difference between the product with heuristic and tailored cooling systems' experimental cell temperature results.

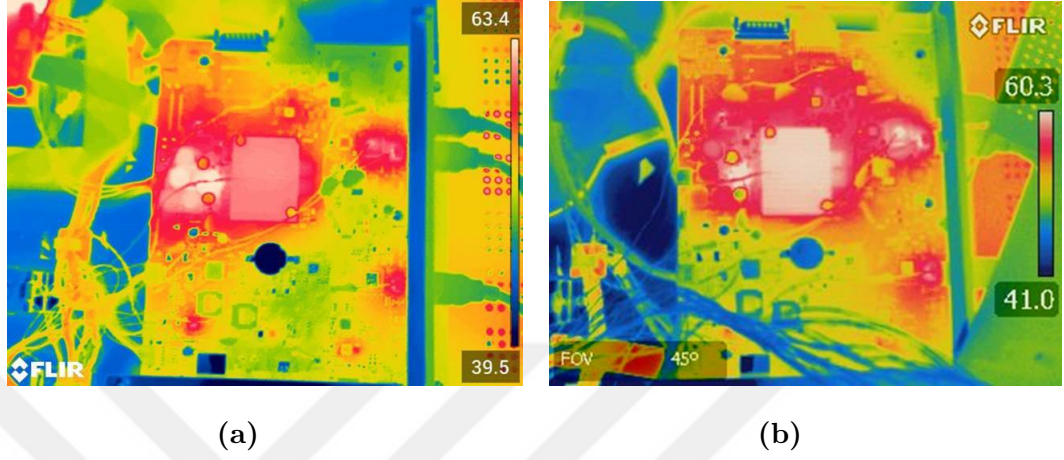
| MEASUREMENT POINTS | HEURISTIC CELL TEMPERATURES (°C) | TAILORED CELL TEMPERATURES (°C) | DIFFERENCE (°C) |
|--------------------|----------------------------------|---------------------------------|-----------------|
| 1                  | 51.3                             | 51.3                            | 0.0             |
| 2                  | 52.1                             | 52.3                            | 0.2             |
| 3                  | 49.5                             | 50.3                            | 0.8             |
| 4                  | 50.5                             | 52.4                            | 1.9             |
| 5                  | 51.7                             | 51.8                            | 0.1             |
| 6                  | 50.1                             | 50.1                            | 0.0             |
| 7                  | 50.5                             | 51.5                            | 1.0             |
| 8                  | 50.4                             | 53.3                            | 2.9             |
| 9                  | 51.3                             | 53.4                            | 2.1             |
| 10                 | 52.4                             | 53.1                            | 0.7             |

From Table 13, the detailed temperature difference between the same points of the cells of the heuristic and tailored cooling systems can be seen, and it is negligible. For both cases, the temperature limits of the cell were met, which was 68°C, according to cell specifications determined by the manufacturer. Figure 47 shows the overall status of the cell temperatures concerning the specification limits.



**Figure 47:** Cell temperatures of the products with heuristic and tailored cooling systems and the cell temperature limit comparison.

Figure 48 and Table 14 show the temperature distribution of MB components with the heuristic and tailored cooling system.



**Figure 48:** Thermal image of the temperature distribution of the MB. (a) Heuristic cooling system. (b) Tailored cooling system.

The figure shows close temperature distributions on the MB for both products. Heat is spotted around the heatsink, which is related to the higher power consumption of the main ICs compared to other components on MBs.

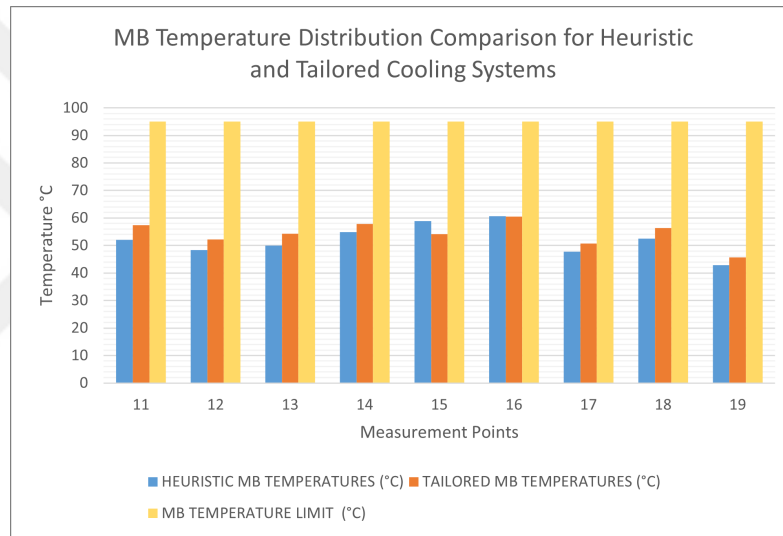
**Table 14:** The difference between the product with heuristic and tailored cooling systems' experimental component surface temperatures of MB results.

| MEASUREMENT<br>POINTS | HEURISTIC<br>MB<br>TEMPERATURES<br>(°C) | TAILORED<br>MB<br>TEMPERATURES<br>(°C) | DIFFERENCE<br>(°C) |
|-----------------------|-----------------------------------------|----------------------------------------|--------------------|
| 11                    | 52.1                                    | 57.4                                   | 5.3                |
| 12                    | 48.3                                    | 52.2                                   | 3.9                |
| 13                    | 49.9                                    | 54.2                                   | 4.3                |
| 14                    | 54.9                                    | 57.8                                   | 2.9                |
| 15                    | 58.9                                    | 54.1                                   | -4.8               |
| 16                    | 60.7                                    | 60.5                                   | -0.2               |
| 17                    | 47.7                                    | 50.7                                   | 3.0                |
| 18                    | 52.5                                    | 56.4                                   | 3.9                |
| 19                    | 42.9                                    | 45.6                                   | 2.7                |

Table 14 shows the detailed temperature difference for the components of MB for

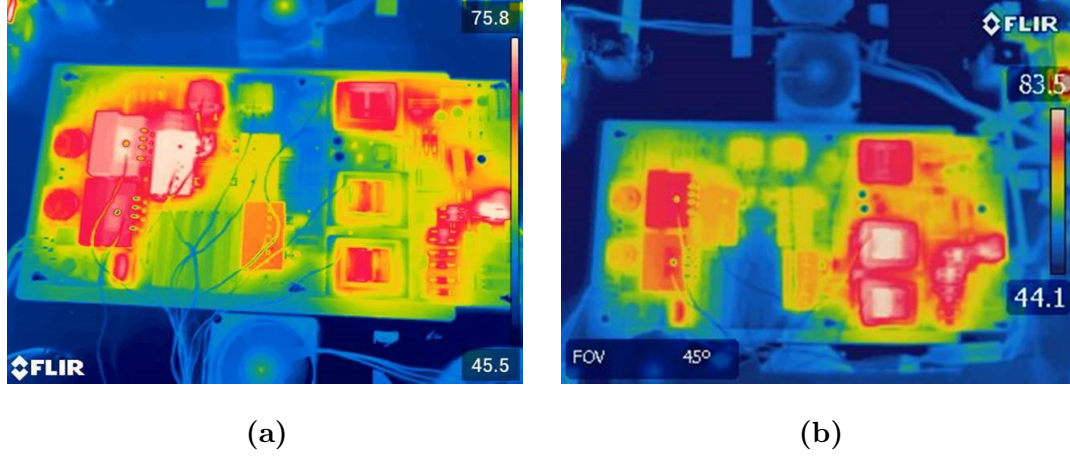


the products with both heuristic and tailored cooling systems, which are negligible. The negative temperature difference indicates that the cooling performance is better for the tailored cooling system, and some components of MB have better temperature levels with the tailored cooling system than the heuristic cooling system. For both cases, the temperature limits of the MB components were met, mainly around 95°C according to component specifications determined by the manufacturer. Figure 49 shows the status of the MB component temperature limits.



**Figure 49:** MB temperatures of the products with heuristic and tailored cooling systems and the MB temperature limit comparison.

Figure 50 and Table 15 show the temperature distribution of PB components with the heuristic and tailored cooling system.



**Figure 50:** Thermal image of the temperature distribution of the PB. (a) Heuristic cooling system. (b) Tailored cooling system.

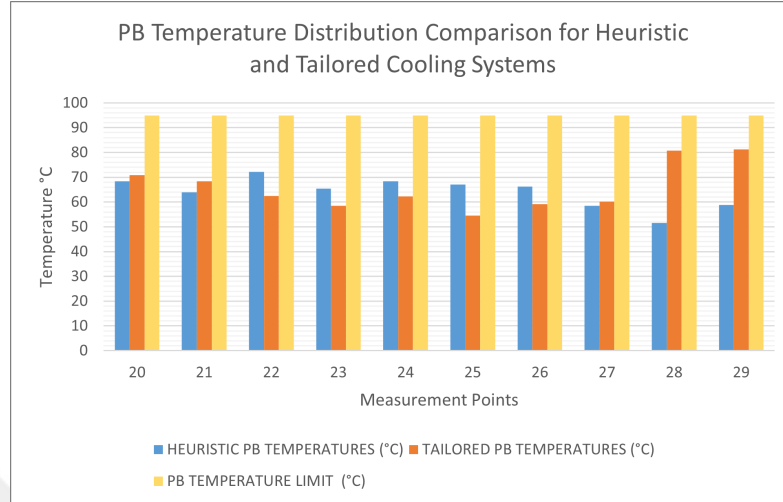
The figure shows close temperature distributions on the PB for both products. Heat is spotted around the transformers and metal oxide semiconductor field-effect transistors (MOSFET); in Figure 50a, heat is spotted around the MOSFETs on the PB of the product with the heuristic cooling system, and this is related to the flow curtain at the center of the PB due to the fan flow. This flow divides the PCB and blocks the heat dissipation of the cooler sides of the PB. In Figure 50b, heat is spotted around transformers on the PB of the product with the tailored cooling system. This is related to the difference between the high power consumption of the transformers compared to MOSFETs.

**Table 15:** The difference between the product with heuristic and tailored cooling systems' experimental component surface temperatures of PB results.

| MEASUREMENT<br>POINTS | HEURISTIC<br>PB<br>TEMPERATURES<br>(°C) | TAILORED<br>PB<br>TEMPERATURES<br>(°C) | DIFFERENCE<br>(°C) |
|-----------------------|-----------------------------------------|----------------------------------------|--------------------|
| 20                    | 68.3                                    | 70.8                                   | 2.5                |
| 21                    | 63.9                                    | 68.3                                   | 4.4                |
| 22                    | 72.1                                    | 62.4                                   | -9.7               |
| 23                    | 65.4                                    | 58.4                                   | -7.0               |
| 24                    | 68.4                                    | 62.2                                   | -6.2               |
| 25                    | 67.1                                    | 54.6                                   | -12.5              |
| 26                    | 66.3                                    | 59.1                                   | -7.2               |
| 27                    | 58.5                                    | 60.1                                   | 1.6                |
| 28                    | 51.6                                    | 80.7                                   | 29.1               |
| 29                    | 58.8                                    | 81.2                                   | 22.4               |

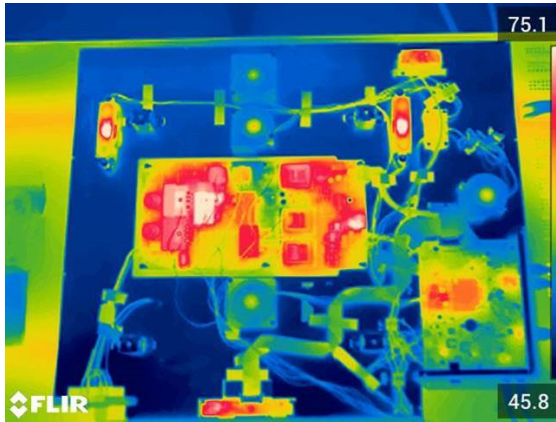
Table 15 shows the detailed temperature difference for the components of PB for the products with both heuristic and tailored cooling systems. Even though most of the temperature differences are negligible, some are major. Measurement points 28 and 29 represent the surface temperature of transformers, and the temperature difference in transformers is relatively high. This is related to the fan positions and the fan flow rates. Transformers can carry relatively high temperatures; however, the temperature of the MOSFETs needs to be controlled for the reliability of the product. With the tailored cooling design, nearly half of the critical components were cooled down with respect to the heuristic cooling system.

For both cases, the temperature limits of the PB components were met, mainly around 95°C according to component specifications determined by the manufacturer. Figure 51 shows the status of the MB component temperature limits.

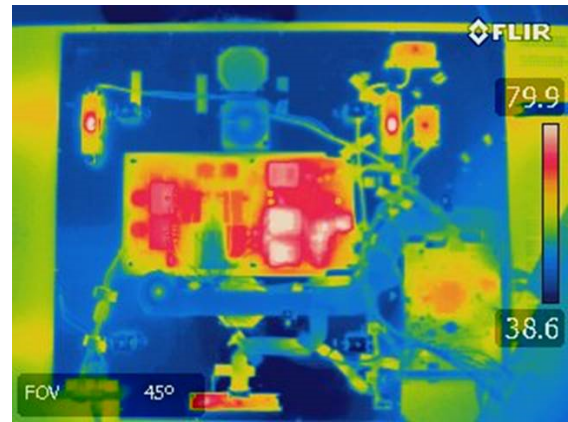


**Figure 51:** PB temperatures of the products with heuristic and tailored cooling systems and the PB temperature limit comparison.

Figure 52 shows the temperature distribution of the back side of the product with the heuristic and tailored cooling systems. The temperature distributions of both systems are similar, as shown in the figure. Adding a fan channel to the system significantly affects the product's cooling performance.



(a)

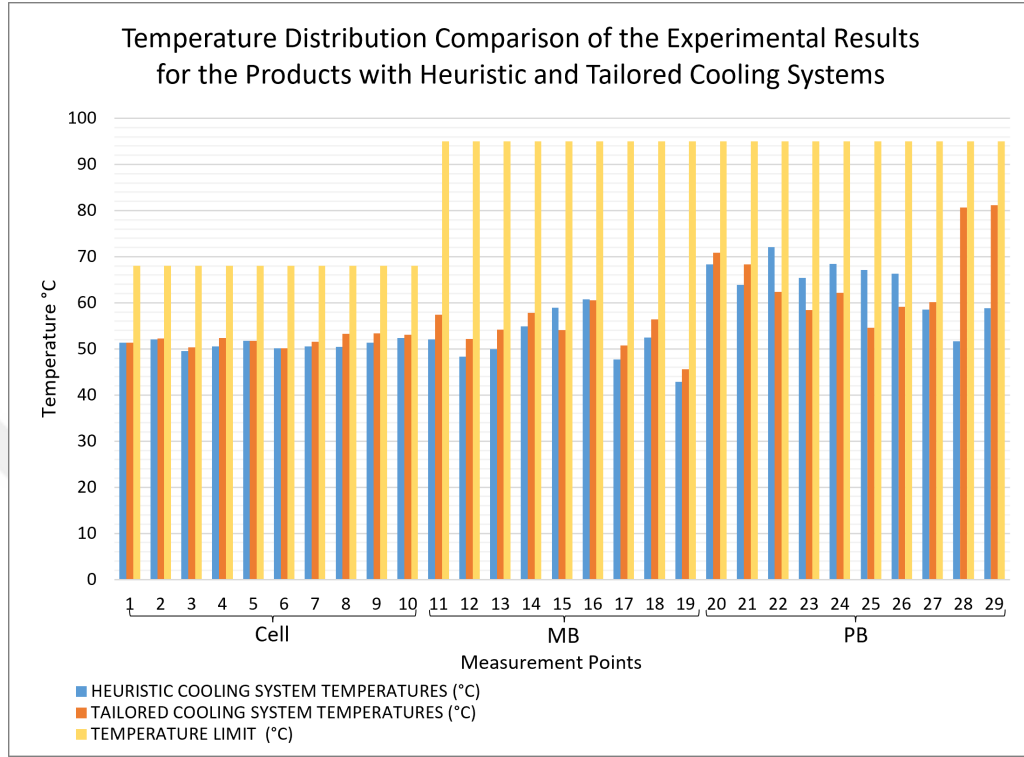


(b)

**Figure 52:** Thermal image of the temperature distribution of the back side of the product. (a) Heuristic cooling system. (b) Tailored cooling system.

Figure 53 shows the cell, MB, and PB temperature comparisons of the products

with a heuristic cooling system and a tailored cooling system.

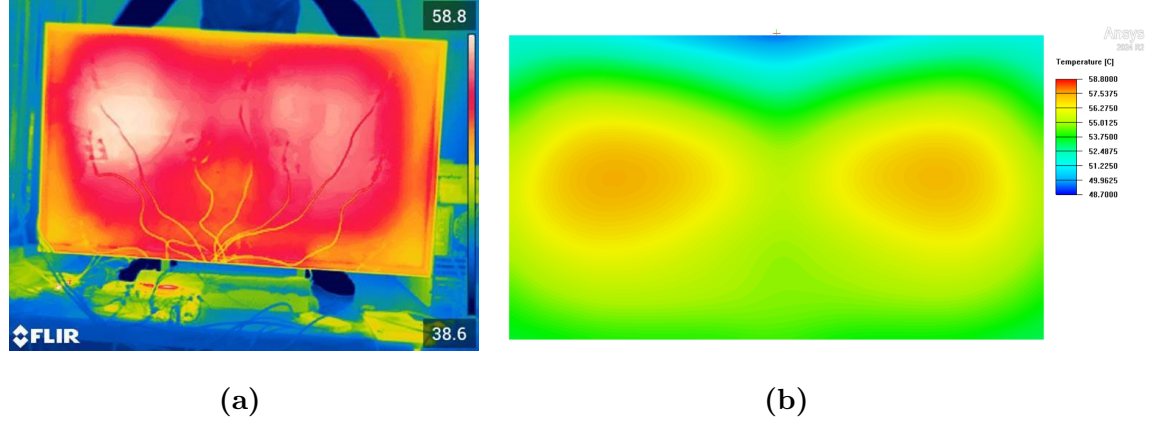


**Figure 53:** Temperature distribution comparison of the experimental results for the products with heuristic and tailored cooling systems.

The temperature differences between heuristic and tailored cooling systems are relatively low. Thus, the tailored cooling system provides a cooling performance comparable to the heuristic cooling system. Additionally, cost savings are achieved by eliminating one axial fan and one radial fan from the cooling system.

#### 4.3.2 Experimental and Numerical Results Comparison of Heuristic Cooling System

Figure 54 is obtained from Figure 46a and Figure 32. It illustrates the comparison of the cell temperatures between the experimental and numerical results of the heuristic cooling system. It also provides an idea of the thermal performance, highlighting the inconsistencies between the experimental data and the numerical simulations.



**Figure 54:** Temperature distribution comparison of the cell for the heuristic cooling system. (a) Experimental temperature distribution. (b) Numerical temperature distribution.

Table 16 is structured by using Table 13 and Table 5. It provides the temperature difference between experimental measurements and numerical results for the cell temperatures of the heuristic cooling system.

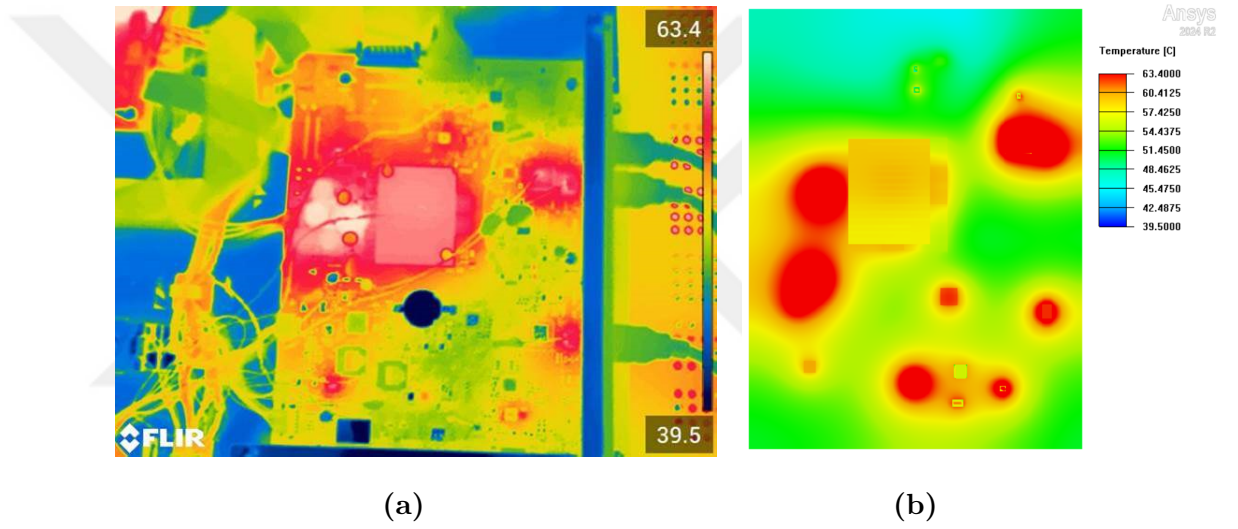
**Table 16:** The difference between experimental and numerical results of the cell temperatures of the heuristic cooling system.

| MEASUREMENT<br>POINTS | NUMERICAL<br>CELL<br>TEMPERATURES<br>(°C) | EXPERIMENTAL<br>CELL<br>TEMPERATURES<br>(°C) | DIFFERENCE<br>(°C) |
|-----------------------|-------------------------------------------|----------------------------------------------|--------------------|
| 1                     | 53.0                                      | 51.3                                         | 1.7                |
| 2                     | 56.6                                      | 52.1                                         | 4.5                |
| 3                     | 54.5                                      | 49.5                                         | 5.0                |
| 4                     | 52.7                                      | 50.5                                         | 2.2                |
| 5                     | 56.4                                      | 51.7                                         | 4.7                |
| 6                     | 54.3                                      | 50.1                                         | 4.2                |
| 7                     | 51.2                                      | 50.5                                         | 0.7                |
| 8                     | 54.7                                      | 50.4                                         | 4.3                |
| 9                     | 57.1                                      | 51.3                                         | 5.8                |
| 10                    | 57.2                                      | 52.4                                         | 4.8                |

The experimental and numerical comparison of the heuristic cooling system shows

differences between simulation and real-life cases. From Figure 54, the temperature distribution of the experiment is similar to the simulation results. Table 16 supports this similarity by the temperature differences. The temperature differences are considered acceptable according to simulation and real-life case comparisons.

Figure 55 is obtained from Figure 48a and Figure 33. It illustrates the comparison of the MB temperatures between the experimental and numerical results of the heuristic cooling system.



**Figure 55:** Temperature distribution comparison of the MB for the heuristic cooling system. (a) Experimental Temperature distribution. (b) Numerical temperature distribution.

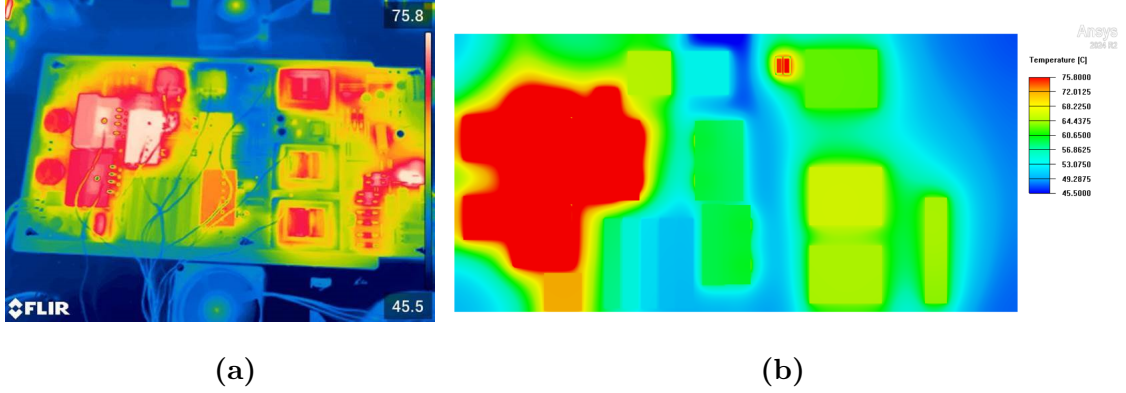
Table 17 is structured by using Table 14 and Table 8. It provides the temperature difference between experimental measurements and numerical results for the MB temperatures of the heuristic cooling system.

**Table 17:** The difference between experimental and numerical results of the MB temperatures of the heuristic cooling system.

| MEASUREMENT<br>POINTS | NUMERICAL<br>MB<br>TEMPERATURES<br>(°C) | EXPERIMENTAL<br>MB<br>TEMPERATURES<br>(°C) | DIFFERENCE<br>(°C) |
|-----------------------|-----------------------------------------|--------------------------------------------|--------------------|
| 11                    | 65.7                                    | 52.1                                       | 13.6               |
| 12                    | 66.2                                    | 48.3                                       | 17.9               |
| 13                    | 62.7                                    | 49.9                                       | 12.8               |
| 14                    | 59.6                                    | 54.9                                       | 4.7                |
| 15                    | 54.0                                    | 58.9                                       | -4.9               |
| 16                    | 71.5                                    | 60.7                                       | 10.8               |
| 17                    | 65.1                                    | 47.7                                       | 17.4               |
| 18                    | 60.4                                    | 52.5                                       | 7.9                |
| 19                    | 58.3                                    | 42.9                                       | 15.4               |

For the MB side, Figure 55 shows the temperature distribution of the experimental and numerical results of the heuristic cooling system. Unlike cell temperature distribution, there are differences between the heat spot locations on the MB. This is related to the theoretical power consumption of the components and the actual power consumption of the components while the system is operating. In addition, PCB modeling in simulations was isotropic, which is one of the causes of the difference. It is also related to the IR camera position while the image is recorded. Table 17 shows the detailed difference between the two cases, and the difference is relatively high for some components. A negative temperature difference means that experimental temperature measurements are higher than the numerical ones. Figure 56 is obtained from Figure 50a and Figure 33. It illustrates the comparison of the PB temperatures between the experimental and numerical results of the tailored cooling system.





**Figure 56:** Temperature distribution comparison of the PB for the heuristic cooling system. (a) Experimental Temperature distribution. (b) Numerical temperature distribution.

Table 18 is structured by using Table 15 and Table 8. It provides the temperature difference between experimental measurements and numerical results for the PB temperatures of the heuristic cooling system.

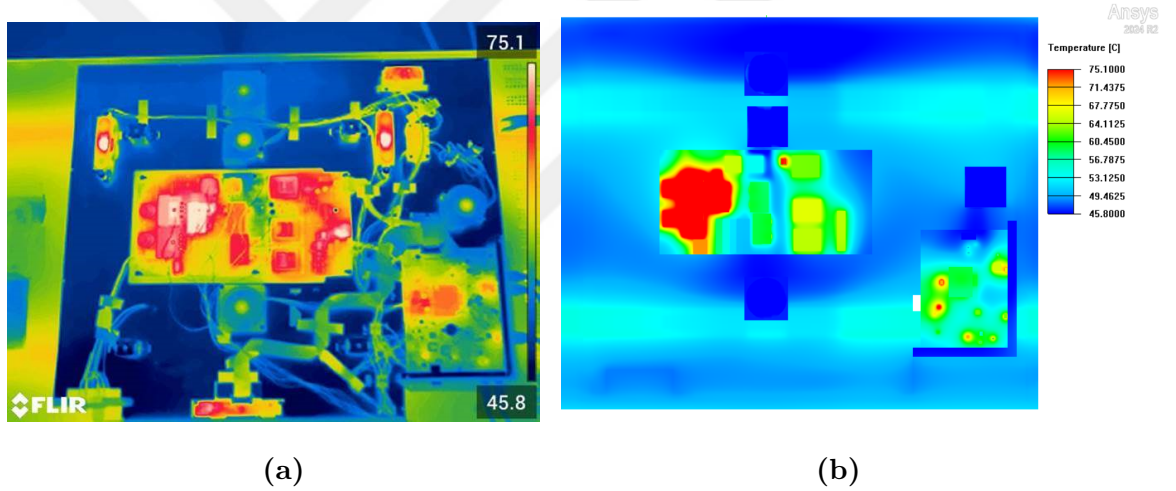
**Table 18:** The difference between experimental and numerical results of the PB temperatures of the heuristic cooling system.

| MEASUREMENT<br>POINTS | NUMERICAL<br>PB<br>TEMPERATURES<br>(°C) | EXPERIMENTAL<br>PB<br>TEMPERATURES<br>(°C) | DIFFERENCE<br>(°C) |
|-----------------------|-----------------------------------------|--------------------------------------------|--------------------|
| 20                    | 91.6                                    | 68.3                                       | 23.3               |
| 21                    | 86.2                                    | 63.9                                       | 22,3               |
| 22                    | 82.5                                    | 72.1                                       | 10.4               |
| 23                    | 59.1                                    | 65.4                                       | -6.3               |
| 24                    | 59.6                                    | 68.4                                       | -8.8               |
| 25                    | 76.0                                    | 67.1                                       | 8.9                |
| 26                    | 65.1                                    | 66.3                                       | -1.2               |
| 27                    | 53.7                                    | 58.5                                       | -4.8               |
| 28                    | 66.4                                    | 51.6                                       | 14.8               |
| 29                    | 64.5                                    | 58.8                                       | 5.7                |

For the PB side, the errors are relatively high, the same as those on the MB side.

From Figure 56 and Table 18, it can be said that the actual power consumption of the components is different from the simulations, which is related to assumptions and the actual power consumption differences due to the PB PCB layout.

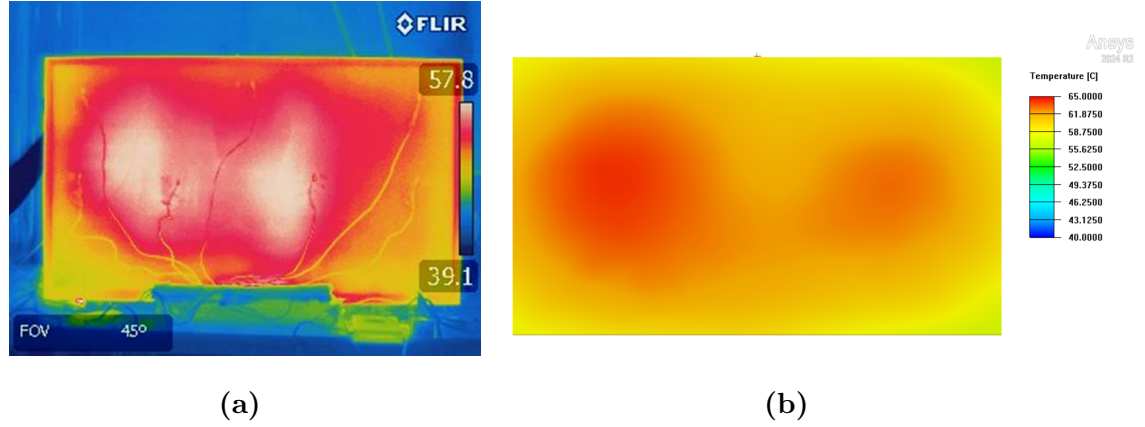
Figure 57 compares the experimental and numerical temperature distribution of the back side of the heuristic cooling system. The temperature distribution of the back side of the experiments and numerical results are close to each other except for some components on MB and PB represented by Figure 57. The assumptions, theoretical and practical power consumption differences, and model simplifications for the simulations cause these differences.



**Figure 57:** Temperature distribution comparison for the back side of the product with the heuristic cooling system. (a) Experimental Temperature distribution. (b) Numerical temperature.

#### 4.3.3 Experimental and Numerical Results Comparison of Tailored Cooling System

Figure 58 is obtained from Figure 46b and Figure 36. It illustrates the comparison of the cell temperatures between the experimental and numerical results of the tailored cooling system. It also provides an idea of the thermal performance, highlighting the inconsistencies between the experimental data and the numerical simulations.



**Figure 58:** Temperature distribution comparison of the cell for the tailored cooling system. (a) Experimental Temperature distribution. (b) Numerical temperature distribution.

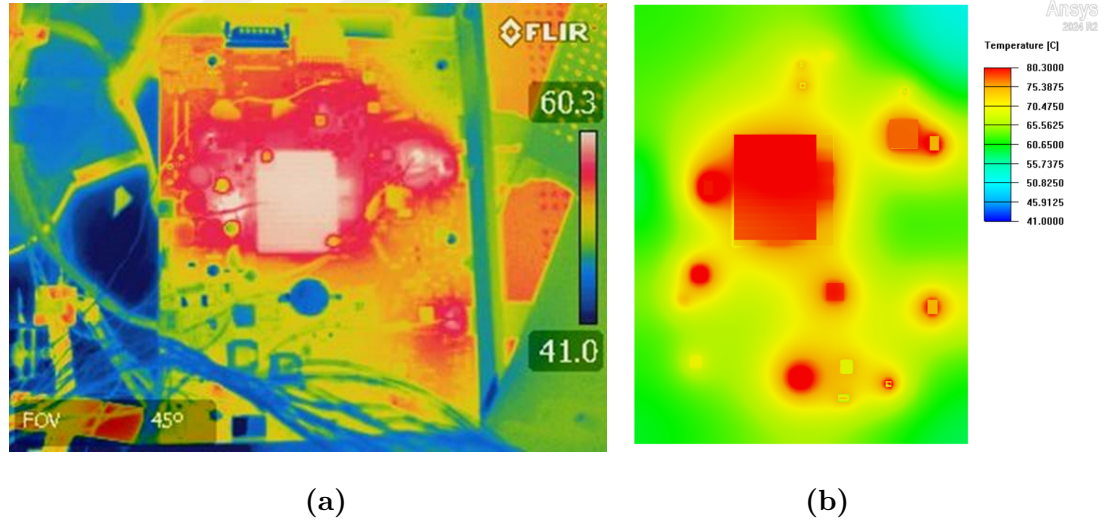
Table 19 is structured by using Table 13 and Table 7. It provides the temperature difference between experimental measurements and numerical results for the cell temperatures of the tailored cooling system.

**Table 19:** The difference between experimental and numerical results of the cell temperatures of tailored cooling system.

| MEASUREMENT<br>POINTS | NUMERICAL<br>CELL<br>TEMPERATURES<br>(°C) | EXPERIMENTAL<br>CELL<br>TEMPERATURES<br>(°C) | DIFFERENCE<br>(°C) |
|-----------------------|-------------------------------------------|----------------------------------------------|--------------------|
| 1                     | 60.5                                      | 51.3                                         | 9.2                |
| 2                     | 63.1                                      | 52.3                                         | 10.8               |
| 3                     | 60.9                                      | 50.3                                         | 10.6               |
| 4                     | 59.5                                      | 52.4                                         | 7.1                |
| 5                     | 61.9                                      | 51.8                                         | 10.1               |
| 6                     | 59.5                                      | 50.1                                         | 9.4                |
| 7                     | 60.4                                      | 51.5                                         | 8.9                |
| 8                     | 61.2                                      | 53.3                                         | 7.9                |
| 9                     | 63.9                                      | 53.4                                         | 10.5               |
| 10                    | 62.5                                      | 53.1                                         | 9.4                |

The experimental and numerical comparison of the tailored cooling system shows

the same outcomes as the heuristic cooling system results. Figure 58 shows the temperature distribution of the experiment and the simulations. When the legend of both results is adjusted between the same interval, no temperature variation is observed for the numerical results, which means that the temperature of the cell surface in numerical results is greater than that of the experimental results. That is why the legend of the numerical result is adjusted to the suitable interval for observing the temperature gradient. Table 19 presents the detailed temperature difference between the two results. The differences are relatively high. The reason for the temperature difference may related to the material models for the cell and the simplifications in the model for the simulations. Figure 59 is obtained from Figure 48b and Figure 37. It illustrates the comparison of the MB temperatures between the experimental and numerical results of the tailored cooling system. simulations.



**Figure 59:** Temperature distribution comparison of the MB for the tailored cooling system. (a) Experimental Temperature distribution. (b) Numerical temperature distribution.

Table 20 is structured by using Table 14 and Table 8. It provides the temperature difference between experimental measurements and numerical results for the MB

temperatures of the tailored cooling system.

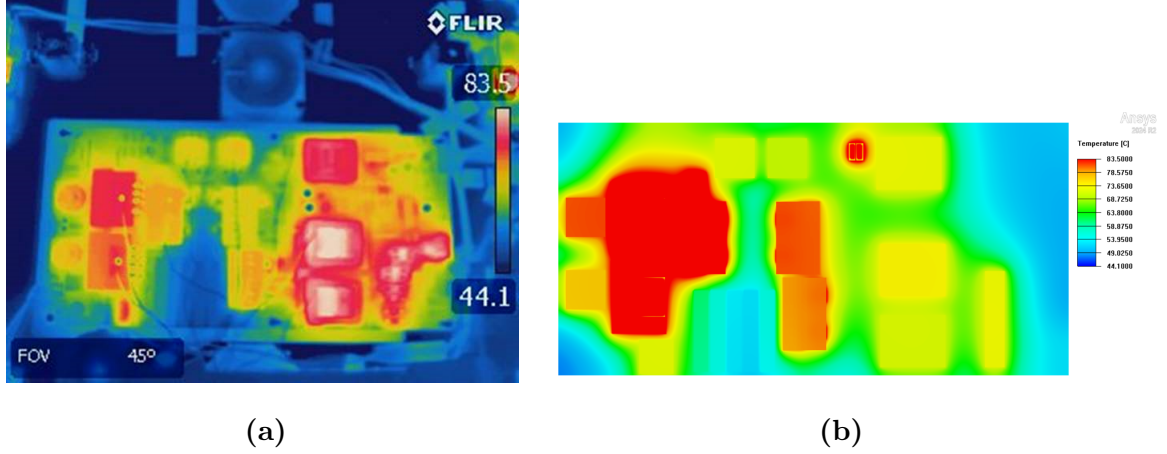
**Table 20:** The difference between experimental and numerical results of the MB temperatures of tailored cooling system

| MEASUREMENT<br>POINTS | NUMERICAL<br>MB<br>TEMPERATURES<br>(°C) | EXPERIMENTAL<br>MB<br>TEMPERATURES<br>(°C) | DIFFERENCE<br>(°C) |
|-----------------------|-----------------------------------------|--------------------------------------------|--------------------|
| 11                    | 77.4                                    | 57.4                                       | 20.0               |
| 12                    | 76.7                                    | 52.2                                       | 24.5               |
| 13                    | 74.7                                    | 54.2                                       | 20.5               |
| 14                    | 80.6                                    | 57.8                                       | 22.8               |
| 15                    | 73.2                                    | 54.1                                       | 19.1               |
| 16                    | 79.8                                    | 60.5                                       | 19.3               |
| 17                    | 80.8                                    | 50.7                                       | 30.1               |
| 18                    | 69.9                                    | 56.4                                       | 13.5               |
| 19                    | 70.9                                    | 45.6                                       | 25.3               |

From Figure 59, the locations of heat spots on MB are similar to experiment and simulation results. However, some regions on MB in simulation have different temperature variations from the experimental results for tailored cooling systems.

Table 20 describes the detailed temperature differences between the same measurement points of numerical and experimental results of MB. The temperature differences are mainly caused by the simplification of the model for the simulation, PCB modeling in the simulations, and the difference between the theoretical and practical power consumptions of the components.

Figure 60 is obtained from Figure 50b and Figure 37. It illustrates the comparison of the PB temperatures between the experimental and numerical results of the tailored cooling system.



**Figure 60:** Temperature distribution comparison of the PB for the tailored cooling system. (a) Experimental Temperature distribution. (b) Numerical temperature distribution.

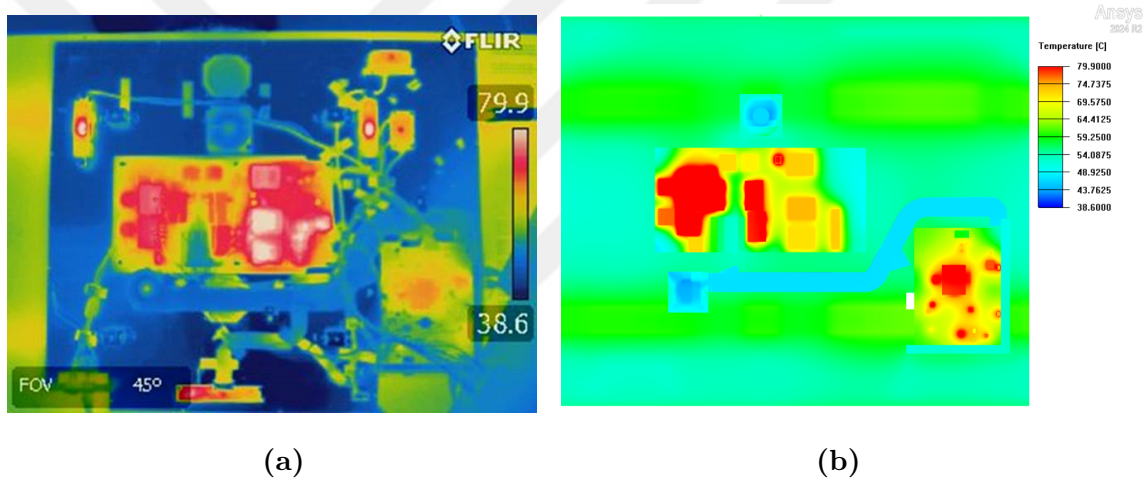
Table 21 is structured by using Table 15 and Table 8. It provides the temperature difference between experimental measurements and numerical results for the PB temperatures of the tailored cooling system.

**Table 21:** The difference between experimental and numerical results of the PB temperatures of tailored cooling system.

| MEASUREMENT<br>POINTS | NUMERICAL<br>PB<br>TEMPERATURES<br>(°C) | EXPERIMENTAL<br>PB<br>TEMPERATURES<br>(°C) | DIFFERENCE<br>(°C) |
|-----------------------|-----------------------------------------|--------------------------------------------|--------------------|
| 20                    | 94.6                                    | 70.8                                       | 23.8               |
| 21                    | 88.6                                    | 68.3                                       | 20.3               |
| 22                    | 91.5                                    | 62.4                                       | 29.1               |
| 23                    | 81.3                                    | 58.4                                       | 22.9               |
| 24                    | 79.8                                    | 62.2                                       | 17.6               |
| 25                    | 90.4                                    | 54.6                                       | 35.8               |
| 26                    | 71.9                                    | 59.1                                       | 12.8               |
| 27                    | 70.3                                    | 60.1                                       | 10.2               |
| 28                    | 75.2                                    | 80.7                                       | -5.5               |
| 29                    | 74.3                                    | 81.2                                       | -6.9               |

Figure 60 and Table 21 have the same outcome as in the heuristic cooling system results. There are temperature differences between some components of the PB. The heat spots of the two cases are at different locations of the PB. This is related to the difference between the actual and theoretical power consumption of the components on PB.

Figure 61 compares the experimental and numerical temperature distribution of the back side of the tailored cooling system, and the experiments and numerical results are different, especially for the MB and PB sides.



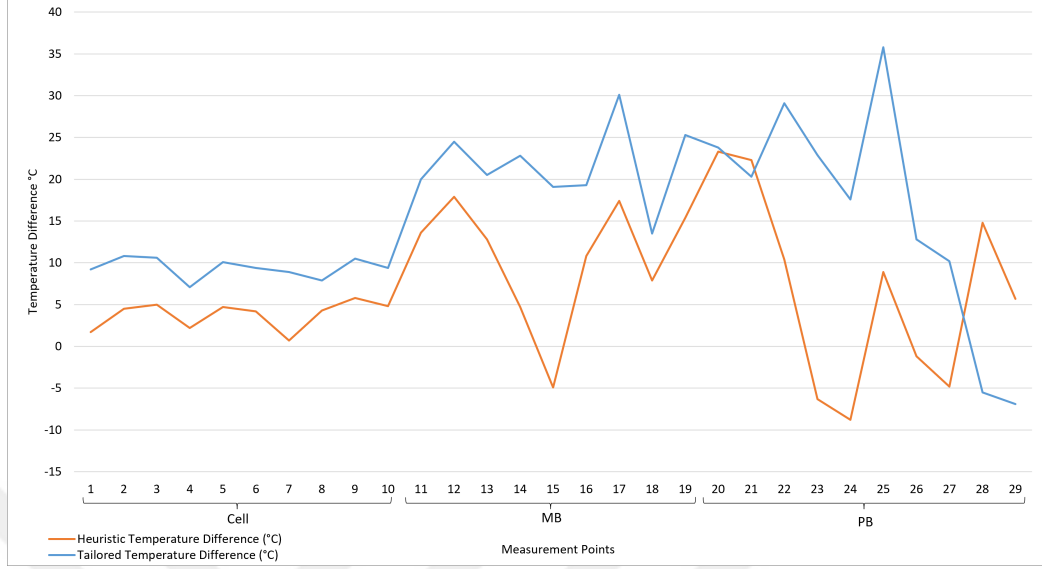
**Figure 61:** Temperature distribution comparison for the back side of the product with the tailored cooling system. (a) Experimental Temperature distribution. (b) Numerical temperature.

Figure 62 compares the temperature deviation between numerical and experimental temperature measurement results for the products with heuristic and tailored cooling systems by subtracting experimental temperature results from numerical temperature results. From the figure, the heuristic and tailored cooling systems demonstrate analogous temperature trends across different measurement points, particularly within the cell surface. Both curves exhibit fluctuations, with the tailored cooling system consistently presenting higher variability. In the cell section (points 1–10), both



systems maintain relatively constant temperature differences, with the tailored cooling system generally demonstrating a higher temperature difference than the heuristic system. The tailored cooling system displays significant temperature fluctuations across the MB and PB areas, with temperature peaks observed at specific measuring points, such as 16, 24, and 28. These variations suggest that the tailored cooling system is particularly responsive or sensitive to individual components, resulting in pronounced localized differences in temperature performance. The heuristic cooling system, in contrast, presents a more consistent and smoother temperature profile across the entire system, with temperature differences remaining relatively low and stable, except for isolated spikes at measurement points 15, 17, and 25. This indicates that the heuristic system may deliver a more uniform cooling effect, although potentially less targeted toward specific components than the tailored system. The comparison between the two cooling systems reveals notable differences in the MB and PB regions. The tailored cooling system demonstrates significantly greater peaks and troughs than the heuristic system. This indicates that the tailored cooling system adapts cooling mechanisms in response to more localized heat loads, resulting in substantial temperature differences. The tailored cooling system demonstrates a potentially more aggressive and targeted approach to cooling, as indicated by substantial variability and more pronounced temperature differences in specific system regions. This characteristic proves advantageous for critical components but also results in more drastic temperature fluctuations. Conversely, the heuristic cooling system represents a more stable and consistent temperature variance, potentially ensuring uniform cooling across the system. However, the heuristic cooling system is less effective in cooling critical components.





**Figure 62:** The comparison of the deviation between numerical and experimental temperatures of the products with heuristic and tailored cooling systems.

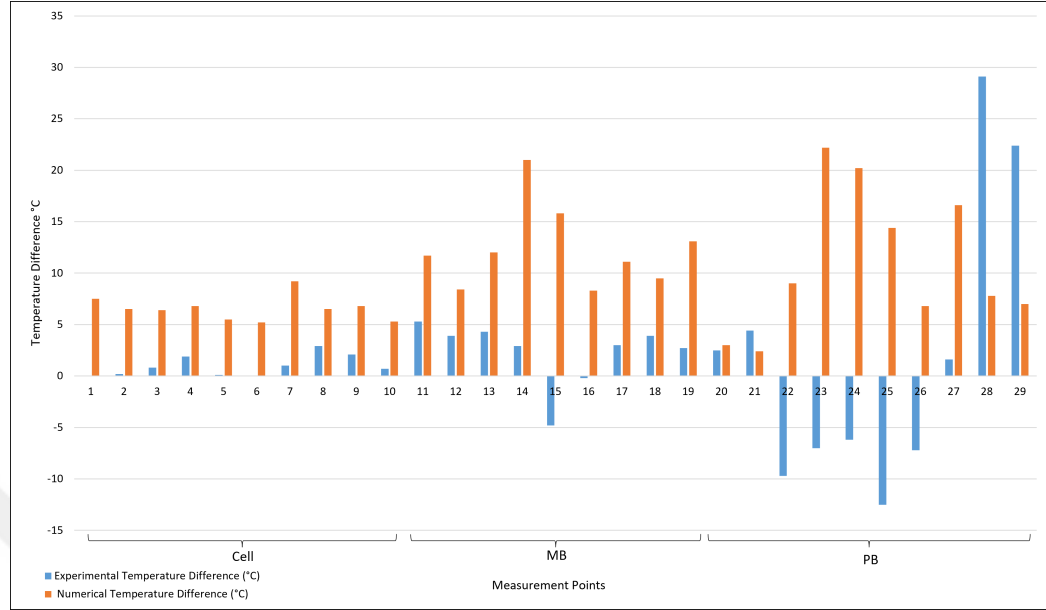
Figure 63 compares the deviation between the numerical results of the products with heuristic and tailored cooling systems and the experimental results of the products with heuristic and tailored cooling systems by subtracting temperature results of the heuristic cooling system from temperature results of the tailored cooling system. Equation 25 explains the calculation of the temperature difference.

$$\begin{aligned} \text{Temperature Difference} = & \text{Temperatures of Tailored Cooling Design} \\ & - \text{Temperature of Heuristic Cooling Design} \end{aligned} \quad (25)$$

From Equation 25, negative temperature differences indicate that the tailored cooling system performs better than the heuristic cooling system, while positive temperature differences indicate the opposite.

The numerical and experimental temperature deviations for the cell show relatively stable temperature differences across the measurement points. The numerical differences are higher than the experimental differences, with the numerical data showing temperature differences around 5°C to 10°C, while the experimental differences

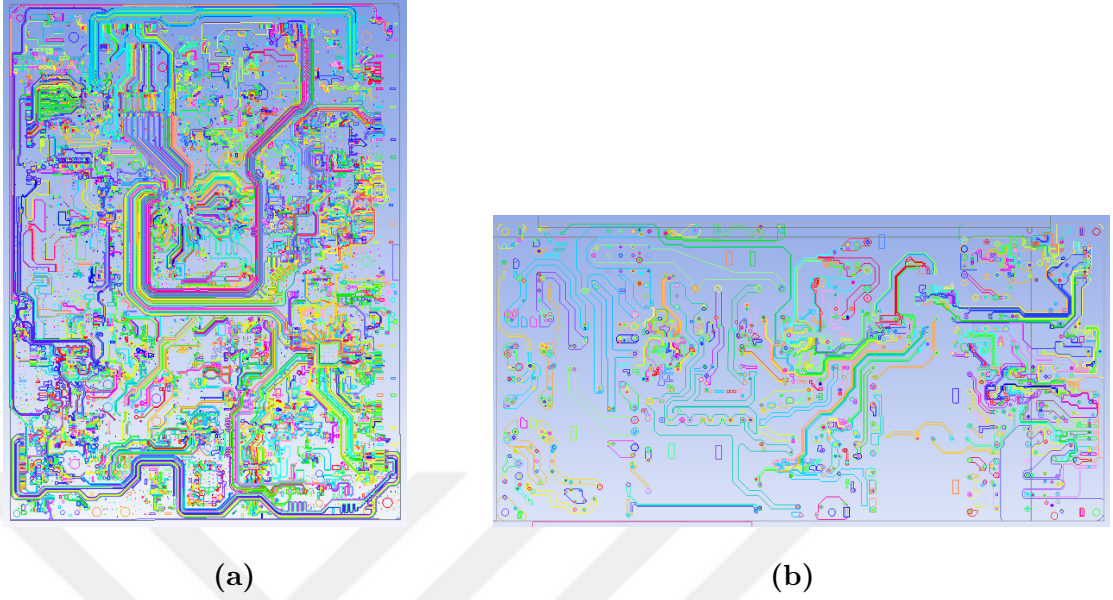
remain close to  $0^{\circ}\text{C}$ . For the MB region, the numerical differences exhibit apparent variability, with peaks at points 14 and 18, reaching differences of  $15^{\circ}\text{C}$  to  $20^{\circ}\text{C}$ . The experimental deviations exhibit fluctuations but remain lower, sometimes dropping below  $0^{\circ}\text{C}$ . The differences between numerical and experimental data increase in this region, indicating that the numerical model gives higher temperature results for some components of the MB. Similar to the MB side, the numerical and experimental results in the PB region are different. The numerical data shows large temperature difference fluctuations, peaking at  $20^{\circ}\text{C}$  to  $30^{\circ}\text{C}$ , while the experimental results show some negative values (as low as  $-10^{\circ}\text{C}$  to  $-15^{\circ}\text{C}$ ). This suggests that the numerical model does not fully capture the cooling dynamics in the PB region. The numerical data tends to overestimate the temperature differences across the system. The higher temperature difference fluctuations in the numerical results, especially in the MB and PB regions, suggest that the model is more sensitive to the cooling dynamics in these areas. The peaks in the numerical results are due to certain assumptions or simplifications in the model that do not account for all the real-world variables present in the experimental setup. The experimental data is generally more stable and shows lower temperature differences across the system. The temperature deviations lower than  $0^{\circ}\text{C}$  in the PB region indicate that the actual cooling performance has been better than predicted in some areas. The lower variability in the experimental data suggests that the actual system is more efficient or uniformly cooled than the numerical model suggests. The numerical and experimental data differences indicate that the numerical model requires further refinement to better align with the experimental results, especially in the PB region. The experimental setup has achieved more stable and efficient cooling, particularly in areas where the numerical model predicted higher temperature differences. This comparison highlights the importance of validating numerical models with experimental data to ensure accurate system performance predictions.



**Figure 63:** The comparison of the deviation between the numerical results of the products with heuristic and tailored cooling systems and the deviation between experimental results of the products with heuristic and tailored cooling systems.

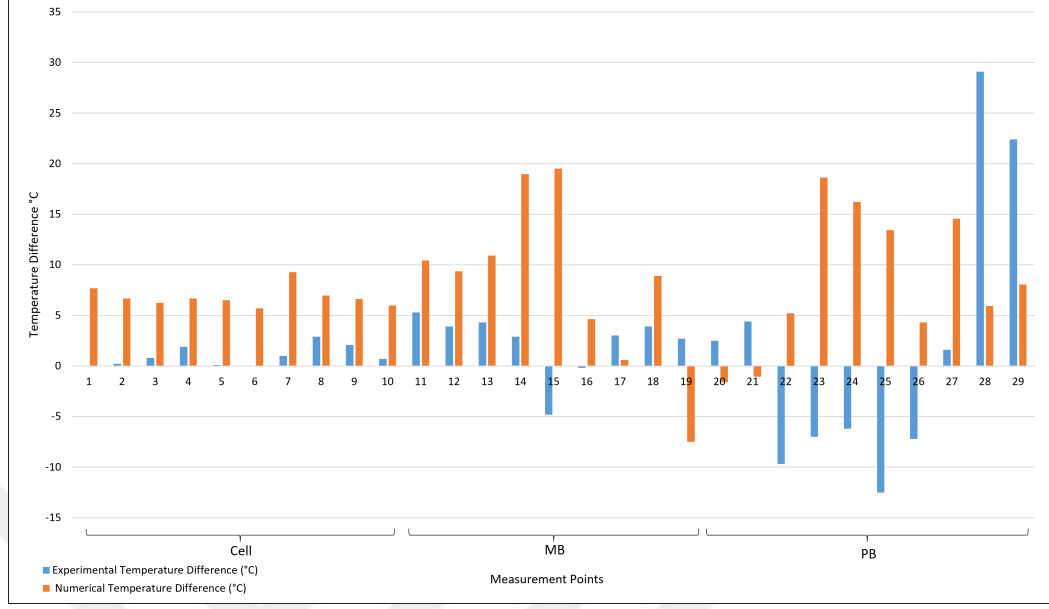
#### 4.3.4 Numerical Refinement Study of the Products with Tailored and Heuristic Cooling Systems

As explained in the previous chapter, many reasons cause differences in the temperatures of products with heuristic and tailored cooling systems. Detailing the material properties of PCB and refining the mesh of the critical components of MB and PB affected the temperature results. By Electronic Computer-Aided Design (ECAD) import, detailed trace modeling of PCB was uploaded to the simulations. This allows the PCB material to be modeled in detail. Figure 64 shows traces and vias of the actual PCB designs of MB and PB.



**Figure 64:** Detailed trace modelling of MB (a) and PB (b) after the ECAD import.

After the ECAD import for the PCB materials, simulations for the product with heuristic and tailored cooling systems were completed. Figure 63 compares the deviation between the numerical results of the products with heuristic and tailored cooling systems and the experimental results of the products with heuristic and tailored cooling systems by subtracting temperature results of the heuristic cooling system from temperature results of the tailored cooling system. Integrating the ECAD file into the simulation for PCB modeling has notably reduced the temperature difference between the numerical and experimental results. Figure 63 shows improved alignment between the numerical and experimental data, especially in the MB and PB regions. However, some differences in the PB region, such as measurement points 23 and 24, still exist, suggesting the need to address other factors or simplifications in the simulation for closer alignment.



**Figure 65:** The comparison of the deviation between the numerical results of the products with heuristic and tailored cooling systems and the deviation between experimental results of the products with heuristic and tailored cooling systems with ECAD import to the PCBs.

The thermocouple measurement error margin is  $\pm 0.5^{\circ}\text{C}$ . Since all the temperature differences indicated in the figures are above this margin, they are significant and critical to follow. Considering the application limitations, temperature differences above  $10^{\circ}\text{C}$  need to be considered as critical temperature differences and focused on. The PB region should be prioritized for addressing critical differences, as this area shows the most significant deviations between numerical and experimental data.

Integrating the ECAD file into the simulation has improved the alignment between the numerical and experimental results, especially in the MB and PB regions, where temperature differences have been reduced. From Figure 63, for the experimental results, points 15, 22, 23, 24, 25, and 26 represent the temperature reduction in the tailored cooling system. However, for points 28 and 29, the temperature increased for the tailored cooling system. This is under critical limits considering component TDS

files and safety margins. While numerical and experimental temperature differences have been reduced, some differences remain. For example, from Figure 65, the temperature of point 15 increased compared to the results in Figure 63. This might be related to the trace intensity near the component because it affects the thermal conductivity of the PCB in this region. For point 19, the temperature difference is lower than the temperature difference in Figure 63. This is also related to the trace intensity. For points 22,23,24,25,26, the temperature differences were reduced compared to the results in Figure 63. ECAD imports to the simulations affect the simulations for better solutions. These differences suggest that further model refinement might be necessary, especially for critical components in the MB and PB regions where overheating could be risky. It is better to incorporate a safety margin into the design to align the numerical and experimental data with the remaining differences. The areas where experimental results show higher temperature differences should be included in the overall safety design to account for potential overheating, especially in the PB region. Including this safety margin will allow the system to operate safely even in conditions where the actual temperatures might exceed the simulated predictions. For the tests carried out, 80% derating was applied for the maximum temperature limits of the components on MB and PB. While the numerical results have improved and are more in line with the experimental data, some differences in the PB region could lead to potential risks. Based on these numerical results, a safe design can be made. Still, it is vital to incorporate a safety margin to cover the areas where experimental temperatures were higher than expected. Further refinement of the simulation model may also help reduce the remaining differences, ensuring the system remains safe under various operational conditions.

## CHAPTER V

### DISCUSSION

This study achieved similar cooling performance by reducing the number of components in the cooling system of the high-bright semi-outdoor LED display device following a design methodology supported by analytical, numerical, and experimental approaches. In literature, thermal management of LED TVs used in indoor applications is achieved by changing the thermal conductivities of cooling components with the combination of natural convection [18] and outdoor display devices managed thermally using forced convection by an open-loop fan structure [19]. This study also uses an open-loop fan structure supported by a fan channel to direct the flow to the critical components of the MB and PB. The thermal conductivity of the materials and types of fans remained the same during the study. The temperature difference between the experiments and the simulations of the product with a tailored cooling system is relatively high compared to the product with a heuristic cooling system. The assumptions and simplifications of the 3D model may cause the difference between numerical and experimental results. In addition, the difference between the power consumption of the components directly obtained from technical datasheets and actual power consumption has a significant effect on the results. Furthermore, assumptions of the material properties of the system components and the measurement errors in temperatures also affect the results. As explained in Chapter 3, the number of channels in data loggers is limited in the study for temperature measurements in the experiments. That is why the measurement points on cell, MB, and PB are limited. Because of this limitation, the LED temperature measurements could not be performed, and the LED temperatures were just compared in numerical results.

The study does not cover the direct and reflected solar irradiation effects from the ground and the buildings, different novel materials, and cooling solution implementations from the literature. In summary, this study shows that the thermal management of a high-bright LED display device can be achieved at the same level of cooling on the components by reducing the number of cooling elements in the existing cooling system and performing a tailored flow-directed study without changing the properties of the cooling elements.





## CHAPTER VI

### CONCLUSIONS

In conclusion, our research provides valuable information on controlling the thermal performance of high-bright LED displays by giving detailed insights from the thermal management methods in literature and presenting workable solutions for indoor and semi-outdoor applications by making tailored cooling system designs while keeping the elements in the cooling system low compared to current cooling system elements. The thermal performance of the tailored cooling system catches the performance of the heuristic cooling system, and it is proved by first with numerical simulations and then experiments as discussed in the discussion section.

This study contributes to the literature by introducing a tailored cooling system for high-bright LED display devices for indoor and semi-outdoor applications. It highlights how tailored cooling systems can achieve comparable cooling performance to heuristic cooling systems while reducing the number of cooling elements. Furthermore, the lack of a study on system-level thermal management of highly bright LED display devices for indoor and semi-outdoor environments in the literature is addressed. With a tailored cooling system, cooling performance is similar to that of the current cooling system. The average temperature variation in the experiments is around 8%. This is achieved by directing the flow via the fan channel to critical components like ICs and MOSFETs, which are essential for the life and reliability of the LED display devices. The temperature on MOSFETs is reduced by an average of 5.5%.

In addition, the methodology of the study combines analytical, numerical, and

experimental approaches to optimize thermal management in high-bright LED displays, particularly in the context of component-specific cooling. This is valuable for future work on high-bright displays for indoor and semi-outdoor environments.

Future research will concentrate on improving cooling design and investigating cutting-edge materials and technologies to improve thermal performance. Studies will also be conducted on products used in outdoor applications. In this case, solar irradiation will be used for the numerical approaches and experiments.



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## VITA

Ahmet Salih Adalı received a B.Sc. degree in Mechanical Engineering from İstanbul Technical University, Istanbul, Turkey, in 2016, and he is currently pursuing an M.Sc. degree in Mechanical Engineering Department at Özyeğin University under the supervision of Assoc. Prof. Özgür Ertunç.

